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INDUSTRIAL ELECTRICITY AND WIRING

REFRIGERATION

INCLUDING AIR CONDITIONING AND COOLING AND HOUSEHOLD AUTOMATIC REFRIGERATING MACHINES

BY

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PREFACE TO THE SECOND EDITION

Progress in the fields of refrigeration and air conditioning during recent years has made necessary a complete revision of this book. For the reason that the previous edition has been extensively used as a textbook in engineering and agricultural schools, the chapters that deal with the thermodynamic theory of refrigeration have been extended. The needs of practical engineers have, however, been constantly in mind during the preparation of the revision so that the usefulness of the book for reference purposes has not been abridged. The index that has been an important factor in making this book adaptable for ready reference has been improved and extended adequately to cover the subjects that have been added.

Important new features, such as improved designs of household and small commercial refrigerating units, silica-gel refrigerating system, new small-capacity absorption plants, methods of controlling refrigerants, revised data on the properties of refrigerants, production of solid carbon dioxide (dry ice), applications of quick freezing and air conditioning, as well as many improvements in refrigeration services, have made necessary the enlargement of the chapters of the preceding edition and the addition of new ones.

The authors are deeply indebted to those engineers and teachers who by their criticisms and suggestions have assisted in the preparation of this revision. They wish to thank particularly Professor R. L. Daugherty, California Institute of Technology; Professor C. H. Fessenden, University of Michigan; Professor V. L. Maleev, Oklahoma A. and M. College; Professor A. J. Ferretti, Northeastern University; Mr. Clarence Birdseye, General Foods Corporation; Mr. P. S. Staples, Creamery Package Manufacturing Company; Mr. H. F. Ryder, H. P. Hood & Sons; Mr. E. R. Ryan and Mr. W. L. Cummings, Frigidaire Corporation; Mr. Terry Mitchell and Mr. R. S. Zeihms, Frick Company; Mr. P. M. Johnson and Mr. E. H. Whitney, Kelvinator Sales Corporation; Mr. Crosby Field, Flakice Corporation; Mr. A. R. Stevenson, Jr., and Mr. F. C. Sarchet, General Electric Company;

Mr. R. E. Finnin, Boston Consolidated Gas Company; Mr. G. M. Craige of Gentsch & Thompson; Mr. F. W. Robinson, Armstrong Cork and Insulation Company; Mr. Louis S. Davis, York Ice Machinery Co.; and Mr. Thomas Coyle, Roessler and Hasslacher Chemical Company.

Special mention should be made of the contribution of Mr. John E. York, designing engineer of the Stone and Webster Engineering Corporation, for the unusually complete set of practical problems on air conditioning and cooling, as applied to theaters, factories, and office buildings, that are included in the chapter on that subject.

THE AUTHORS.

BOSTON, MASS.
August, 1932.

PREFACE TO THE FIRST EDITION

Refrigeration has come to be an industry of large proportions, and there is a constantly increasing demand for adequate refrigerated storage facilities. The subject is now of great importance not only to operating and designing engineers who have to do with refrigerating plants but also to those who have made a business of the installation and servicing of household refrigerating equipment.

There is an increasing demand for mechanical refrigerating devices which are suitably applicable and safe for producing refrigeration on a small scale in private houses, single apartments, small hotels, restaurants, and stores.

The sphere of usefulness of refrigeration by mechanical means is constantly expanding, and the number of refrigerating machines manufactured is every year larger than in the preceding one. Doubtless, in the near future, practically every house in suburban and urban districts either will be supplied with artificial ice manufactured in large refrigerating plants or will depend for food preservation on refrigerating equipment which is all but completely self-servicing.

Modern refrigeration, which includes practical methods of ice making in a refrigerating plant and of direct cooling without ice, is a comparatively recent development. In the last three-quarters of a century, this industry has expanded to such an extent and our dependence on it has become so complete that our present-day system of freight transportation and of ocean commerce in perishable foods could not exist without it. The feeding of cities, even the avoidance of famine, depends on the facilities for shipping over long distance and for storing in good condition the products of one season for consumption at other times of the year.

Refrigeration is a field which is far from being overcrowded by competent men, and, unquestionably, there are now more new applications of the principles of refrigeration than at any time in the past. There is a new application of the absorption system, for example, in the recently developed gas-heated household refrigerating units.

Unusual features of this book are the data and complete calculations of a commercial test of 15-ton compression refrigerating plant. The index has been prepared carefully and completely so that it may be useful for reference, and that this text-book may also be valuable as an engineers' handbook of information on refrigeration.

To facilitate the rapid correction of problems, teachers using this book as a text may obtain the complete solutions of the problems in the Appendix from the authors upon application.

Representatives of the refrigeration industry have cooperated in many ways in the preparation of this volume by supplying data and by giving valuable suggestions. In this connection, the authors want to mention especially Messrs. Thomas and Raymond T. Shipley, York Manufacturing Company; Mr. G. E. Wallis, Creamery Package Manufacturing Company; Mr. N. H. Hiller, Carbondale Machine Company; Mr. W. H. Carrier, Carrier Engineering Corporation, and Mr. M. J. Nusin, Ingersoll-Rand Company. Mr. John F. Wostrel, Massachusetts Division of University Extension, has contributed many valuable services.

THE AUTHORS.

BOSTON, MASS.
November, 1928.

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REFRIGERATION

CHAPTER I

REFRIGERATION METHODS

Refrigeration, in engineering practice, is a process of removing heat from an enclosed space which is to be maintained at a colder temperature than its surroundings. The idea that refrigeration is a heat-removing process may be more easily understood when one perceives that coldness is really a relative term and that things are hot or cold only as they differ from our everyday experiences. Some degree of heat is, of course, present in all substances at ordinary temperatures.

The earliest method of refrigeration was the cooling of water in *porous* earthenware vessels. By this device, the temperature of the water in the vessel is lowered by the rapid evaporation of the "sweat" which gathers on the surface of the container. In countries where the air is unusually warm and dry, this method can be used with some success. Another mode of cooling which is probably just as old as the evaporative method is to put food and water into a cave or into a stream of flowing water in a place sheltered from the sun. In nearly all countries, there are natural or artificial caves, cellars, and wells (Fig. 1) in which a temperature between 50 and 60° F. may be maintained even in warm weather.

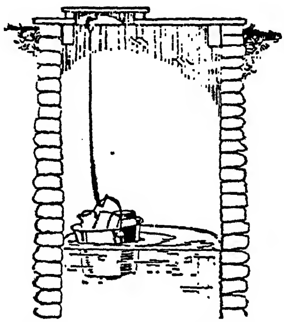


FIG. 1.—Early method of refrigeration.

Ice-box Refrigeration.—The ice box did not come into general use until early in the nineteenth century. Refrigeration is produced in an ice box by the melting of ice in the compartment *A*, as shown in Fig. 2. When the ice melts, a circulation of cold air, as indicated by arrows, is produced in the enclosure, which

keeps foods and liquids cool. For many years, ice-box refrigerators were supplied exclusively with natural ice, that is, with ice which is cut during the winter from the surface of ponds, lakes, or rivers and stored for summer use in buildings called *ice houses*. Natural ice for refrigeration was probably first used in an American home in 1802. A few years later, a shipload of natural ice was sent from Boston to the West Indies. At present, the annual harvest of natural ice in the United States amounts to about 15,000,000 tons which, added to 45,000,000 tons of manu-

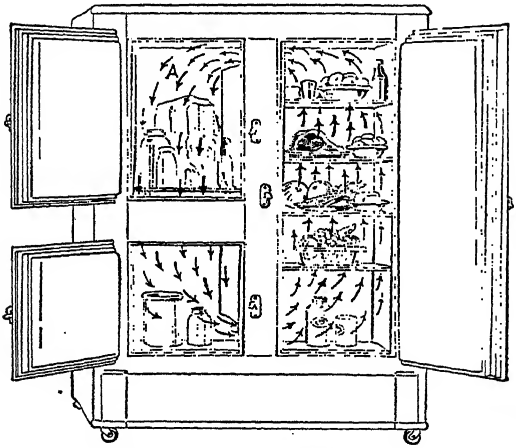


FIG. 2.—Ice-box refrigerati

factured ice, makes the total yearly consumption of ice about 60,000,000 tons.

Heat at Refrigerating Temperatures.—Heat in some amount is present in all substances at ordinary temperatures. It is obvious, of course, that as the temperature of a substance is reduced, there is less heat in it and that with a progressive lowering of temperature, there is always a reduction in the amount of heat down to the absolute zero of temperature, which on the Fahrenheit thermometer is 460° below zero. It has been the experience of those in charge of refrigerating plants that for the proper preservation of food the temperature in the refrigerator should, as a rule, be not much higher than 40° F. that is, 460 + 40 or 500° F. above absolute zero. Thus, there is necessarily a great deal of

heat in the walls, the shelves, the air, and the stored foods in a refrigerator with reference to the absolute zero of temperature. This amount of heat is, in fact, so large that it is possible to boil inside the refrigerator some liquids which have low boiling points, as, for example, liquefied gases and vapors such as liquid air, liquid carbon dioxide, liquid sulphur dioxide, and liquid methyl chloride.

Solids, Liquids, Vapors, and Gases.—It was stated in the preceding paragraph that refrigeration is a method of taking away heat from substances and enclosed spaces. In a study of this subject, it is, therefore, necessary to give some attention to the nature of heat. The accepted theory is that heat springs from the energy of motion of the molecules of which all forms of matter are supposed to be composed. The molecules are in constant motion but are attracted and held together by a force similar to magnetism, which is, however, effective at only short distances. Thus, a substance is said to be *hot*, or *heated*, when its molecules are stirred by some force into violent motion and tend to be driven apart from one another and to be cold when the molecules are relatively inactive and close together. When all heat is removed from a substance so that its temperature is at the absolute zero, the molecules are stationary so that there is, of course, no possibility of the further removal of heat.[1]

Every substance at a given time exists in one of four states or conditions, that is, (1) as a solid, (2) as a liquid, (3) as a vapor, (4) as a gas. These states or conditions differ from each other by the amount of heat that is present, or, in other words, by the amount of movement of the molecules and by the distance between them. In a solid, for example, the molecules are relatively close together, and their movement is consequently so restricted that the substance preserves a definite shape. In a liquid, the molecules are farther apart and have freer movement than in a solid. In a vapor or a gas, the molecular distances and rates of movement are still more increased. In a solid and in a liquid, the molecules tend to hang together, while in a vapor or gas, the molecules tend to get as far apart as possible. Briefly,

[1] By the use of a very carefully constructed refrigerating machine, the investigators at the U. S. Bureau of Standards were able to remove practically all the heat from liquefied hydrogen so that a small portion of it was actually frozen, indicating that the absolute zero of temperature had been nearly reached.

then, it may be stated that all forms of matter exist as solid, liquid, vapor, or gas, according to the amount of heat contained.

Refrigerating Processes.—All refrigerating *processes* depend upon the use of a substance which is readily convertible from a liquid into a vapor, or gas, and also from the vapor or gas into the liquid; and further, these changes must be accomplished within a reasonably narrow range of pressures.

A liquid which boils at a lower temperature than usually prevails inside a refrigerator has the principal requisite for a *refrigerant*, namely, the fluid used as a cooling agent in refrigeration. A small quantity of a liquid refrigerant in a refrigerator, when it boils,[1] will absorb a large amount of heat, which is, of course, its *latent heat* of evaporation.

There is no liquid which in its "natural state" boils at the temperature usual in refrigerators; and, since no such liquids exist in nature, it is necessary to manufacture them. Refrigerating machines are really devices which are used to "manufacture" liquids which will boil at temperatures slightly below those usually required for refrigeration. Such "manufactured" liquids are made from gases or vapors which are *compressed* and then *cooled*. The compression and cooling cause the gas or vapor to liquefy. These "manufactured" gases are expensive, so that it is not economical in the operation of the process of refrigeration to allow them to escape. Nearly all these fluids are, furthermore, objectionable, for various reasons, when discharged into the air. It is, therefore, a part of the process to collect the gases or vapors of refrigerants and bring them back to their liquid state so that they can be used over and over.[2]

[1] The liquid which is ordinarily associated with boiling is water, but water is obviously not suitable for use as a refrigerant, as its boiling point at atmospheric pressure is 212° F. It can be made to boil, however, at a lower temperature than 212° F. by lowering the vapor pressure at its surface as, for example, with a vacuum pump; in fact, one system of refrigeration is based on the boiling of a liquid in a vacuum chamber.

[2] One type of mechanical refrigeration that will be explained later is based on the use of a compressed gas or vapor which is allowed to expand in an engine cylinder and gives up some energy by the loss of heat in proportion to the amount of energy which is converted into work. The refrigerant, in this case, gives up some of its energy in the form of work, as done by the steam engine. The complete process of this kind of refrigeration is not very different from the generation of steam in a boiler and its use in a steam engine, when the steam which is discharged from the engine is condensed and used again.

Natural Ice.—The size of the natural-ice harvest is always uncertain, so that there is a strong incentive for inventors to perfect a mechanical means of refrigeration, in order to make ice from water at any time of the year. From 1830 to 1890, there was relatively little progress in mechanical refrigeration. In 1890, however, when there was an unusually small harvest of natural ice in the United States, the shortage forced an interest in methods and devices for producing ice by mechanical means. In the years immediately following 1890, there was very rapid and successful development of refrigerating machines.

Early History of Refrigerating Machines.—The present systems of mechanical refrigeration depend on the fundamental principle that some vapors or gases which do not ordinarily exist in the liquid state may be liquefied upon being subjected to high pressure. Although this fundamental principle was discovered about 1820, it was not until 1834 that a satisfactory mechanical device was invented to apply this principle to the refrigerating process. The inventor of this device was Jacob Perkins, a Massachusetts mechanician and engineer. This first compression refrigerating machine used ether as a refrigerant. In 1855, the first absorption refrigerating machine (see p. 9), using ammonia as a refrigerant, was produced in Germany.

Refrigeration by Compression.—One of the methods of "manufacturing" a liquid refrigerant which will boil in refrigerators below the usual temperatures requires the use of a compressor which has the effect of increasing both the temperature and the pressure of the vapor or gas of the refrigerant which is used. In other words, in this compression, the vapor or gas of the refrigerant is used as the raw material. The vapor or gas of the refrigerant becomes hot by this compression, and the next step in the process is to cool it in a condenser by the use of a stream of cold water or, in some cases, by cool air. In this part of the process, the molecules of vapor or gas are pushed together so closely by compression and their movement reduced by cooling to such an extent that the attractive force between them becomes effective, and, as the cooling continues, the vapor or gas gradually condenses and becomes liquid. A condenser as used for this purpose consists usually of a metal coil containing the compressed vapor or gas, and the coil is surrounded by cold water or cool air.

As the process is continued, the cool liquid refrigerant, which has been condensed from the vapor or gas, is forced through a

small orifice, usually in the form of an expansion or throttling valve. In this valve, the pressure of the liquid refrigerant is reduced to such an extent that it will boil in the cooling coil of the *evaporator* at a sufficiently low temperature to maintain satisfactory refrigeration. After the liquid refrigerant goes through the expansion valve, it passes on into the cooling coils of the evaporator, consisting usually of a coil of pipes which is connected to the low-pressure or "suction" side of the compressor.

It may be interesting to explain here why the compression of the refrigerant and its subsequent expansion are necessary. The reason for the compression is that when the pressure of the vapor of a refrigerant is increased, the temperature of its boiling point is raised in proportion to the increase of pressure; and, similarly, the reason for the expansion is that when the pressure is reduced, the temperature of the boiling point is lowered.

Applications of Refrigeration.—The compression type of refrigerating system which was invented by Perkins, after further development by other inventors, became a practical machine, and, about 1855, commercial quantities of ice were produced.

The first shipments of refrigerated fruit were made in 1866. These shipments were in large boxes containing 200 quarts of strawberries which were packed with 100 pounds of ice.

In 1872, the first successful shipments of beef and fish were made in railway cars which were heavily insulated on all sides and were refrigerated with natural ice. This was the beginning of the refrigerator-car industry.

Mechanical refrigeration has found application in a number of other industries, as, for example, in the refining of oil, where a refrigeration apparatus is used for the removal of paraffin; also, in the ventilation of buildings in warm weather when a cold liquid is circulated to reduce the temperature of the air used for ventilating. Refrigeration is applied in metallurgical operations, as, for example, in removing the moisture from the air which enters the blast furnaces. Other applications are in the manufacture of textiles, in the curing of tobacco, in the manufacture of cigars, in the making of candy, in the making of photographic films and similar celluloid products, and likewise in surgical operations and in excavating; this last process is accomplished by freezing a ring of quicksand so that tunneling can be done in the difficult material.

CHAPTER II

SYSTEMS OF REFRIGERATION

Simple Refrigeration Devices.—In an ice-box refrigerator, foods are kept cool by the melting of ice. The ice is, of course, the refrigerating medium and serves to remove heat from the foods as well as also to absorb the large amount of heat which enters through the insulation. It must always be kept in mind that the fundamental principle underlying the operation of any refrigerating device is the transfer of heat from one body to another by the method of temperature equalization.[1]

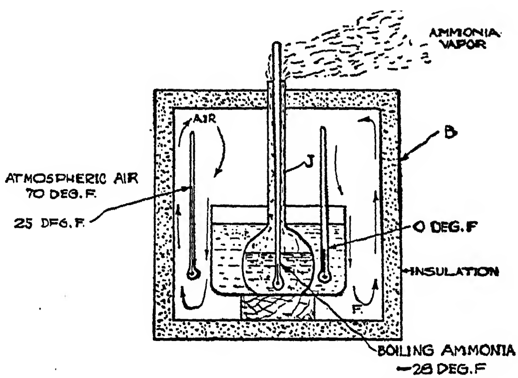


FIG. 3.—Elementary refrigerating apparatus.

Figure 3 shows the application of an elementary evaporating apparatus for producing refrigeration. A jar *J* which is partly filled with liquid ammonia is shown inside a suitable box *B*. The principle of operation is, however, the same if some other refrigerating medium such as sulphur dioxide were used. If the jar is *open at the top*, the temperature of the ammonia vapor which is given off by the liquid will be −28° F., and the insulated

[1] If two bodies have different temperatures and are placed near together, the heat in the hotter body has a tendency to flow into the colder body until the temperatures are the same. By this method, the hotter body is refrigerated, and the colder one is heated. Heat always flows from a hot body into a colder body just as water flows from a high to a lower level.

bodies surrounding the ammonia jar *J* may easily be maintained at a temperature of 0° F. when the temperature outside the box *B* is about 70° F. In the operation of this device, the heat of the air inside the box is absorbed by the ammonia when it evaporates vigorously and boils in the jar. During this evaporating process, it is possible to maintain in the ammonia jar a uniform temperature. In the apparatus shown in Fig. 3, the vapor from the evaporation of the refrigerant escapes to the atmosphere from the top of jar *J*.

Ice-freezing System.—A modification of the last figure is shown in Fig. 4, which is an application of the circulation of cold brine for use in a box or tank in which artificial or manufactured

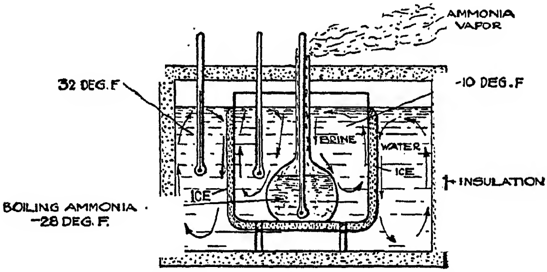


Fig. 4.—Elementary ice-making apparatus.

ice is made. In this figure, there is a jar containing liquid ammonia which is immersed in brine. The brine tank is supported in a larger tank containing the water from which ice is to be made. The jar containing the liquid ammonia has a long, narrow neck which passes up through the top of the larger tank. The pressure in the ammonia jar is, therefore, atmospheric, and the temperature of the ammonia is, consequently, $-28°$ F., which is sufficiently low to maintain a temperature of from 10 to 15° F. in the brine tank. This low temperature of the brine will cause the water surrounding the brine tank to freeze on the surface of the tank. In this freezing process, the ammonia absorbs a quantity of heat from the brine; and the brine, in turn, absorbs, when freezing the water to ice, approximately the same quantity of heat from the water in the outside tank.[1] The brine is made of

[1] In addition to this amount of heat exchange in the actual process of freezing, the brine absorbs also the heat required to cool the water to the freezing point as well as any heat which enters the water through the walls of the tank.

such a concentration that it does not freeze at the temperatures usually maintained in the brine tank, so that there may always a circulation of brine.

Systems of Mechanical Refrigeration.—The of mechanical refrigeration are

a. The air system, in which air is used as the refrigerant and this air is first compressed and is then expanded in very much the same way as the steam in a steam engine, giving up energy to a moving system (similar to an engine) and, in this way, losing heat.

b. The compression system, using ammonia, carbon dioxide, sulphur dioxide, or some refrigerant with similar properties, is so called to distinguish it from a third system (which is mentioned below), because a *compressor* is used to raise the pressure of the vapor of the refrigerant and deliver it to the condenser after removing it from the cooling coils or the evaporator.

c. The absorption ammonia system is so called because a weak ammonia solution removes ammonia vapor from the cooling coils of the evaporator by *absorption* and the richer ammonia solution so formed is then pumped into a high-pressure chamber called a *generator*. By heating the generator, the ammonia vapor is driven off from the liquid and passes through suitable piping into the condenser.

No matter what system is used, a circulating fluid, usually water or air, is employed to carry away the heat, so that the temperature of the cooling fluid limits the liquefying temperature in the system and indirectly limits also the maximum permissible pressure.

Refrigerating Machines Using Air.—The air system has two essential parts: (1) the air compressor and (2) the air engine or expansion motor which operates by the expansion of the high-pressure air which comes from the air compressor. Usually, the air engine or expansion motor is connected mechanically to the shaft driving the compressor so that the work done by the air motor assists in compressing the air which is used in the system. In the operation of this system, the compressor takes in and compresses air to a high pressure. The compression of the air produces heat just as in any other compression system. The compressed air is discharged from the compressor into the coils of the cooler in which heat is removed by cool water which circulates through the coils. The cooled compressed air which is at high pressure is then used to drive the air engine or motor.

In the *expansion of the compressed air* in the air engine or motor, the air becomes very cold and is carried off in pipes to the refrigerating rooms, where it absorbs heat and can finally pass off into the outside air.

The air machines intended for refrigeration purposes have not been applied to any considerable extent in small units, because of the very high cost of equipment, which consists, of course, of two machines, that is, the compressor and the air engine or motor. In comparison, the ammonia compression system of refrigeration has only one machine—the ammonia compressor. From this comparison, it will be seen that the air refrigerating system is likely to cost about twice as much as the ammonia compression system and may probably cost twice as much for repairs because of the multiplicity of moving parts, all of which must operate at high pressures. There is also likely to be considerable trouble caused by the freezing of moisture carried into the compressor with the atmospheric air. This difficulty may, however, be eliminated by the use of a *dense air refrigerating system*[1] in which the same air is used over and over again without discharging any into the outside air. The dense air machine for refrigeration is explained on page 260.

Compression System of Refrigeration.—Figure 5 shows a simple form of the compression system of refrigeration. In the operation of this system, there is alternating compression and expansion of the refrigerant. The object of compressing the vapor of the refrigerant is to increase its boiling point, because, as the pressure of a vapor is increased, the temperature of its boiling point is also raised. Similarly, the reason for expanding the refrigerant is that when the pressure is reduced, the temperature of the boiling point is also lowered.[2] In the figure, the essential parts of a compression refrigerating system are shown,

[1] This system is so called because in order to reduce the size of the cylinders and pipes through which the air circulates, its pressure is never permitted to get so low as atmospheric.

[2] If a refrigerant in the liquid state is brought into a room where the temperature is higher than the boiling point of the refrigerant, its temperature will rise until the boiling point is reached, when it will evaporate or boil at a constant temperature depending on the pressure. Similarly, if a refrigerant in the vapor state is brought into a room where the temperature is lower than the condensing point of the vapor of the refrigerant, its temperature will be lowered until the condensing point is reached, and at this temperature, which will remain constant, all the vapor will be condensed.

and the typical temperatures are given for refrigeration *with ammonia.* The working cylinder C of the compressor as shown at the top of the figure has two valves, one S for the suction, and the other D for the discharge of the compressed vapor. In the operation of the compressor, the piston first reduces the pressure in the cylinder C somewhat below the pressure in the *evaporator,* which is shown at the right-hand side of the figure. This reduction of pressure in the evaporator causes the vapor of the refrigerant to flow through the suction valve into the cylinder of the compressor. The pressure in the evaporator is determined largely by the temperature which is required for the refrigerating

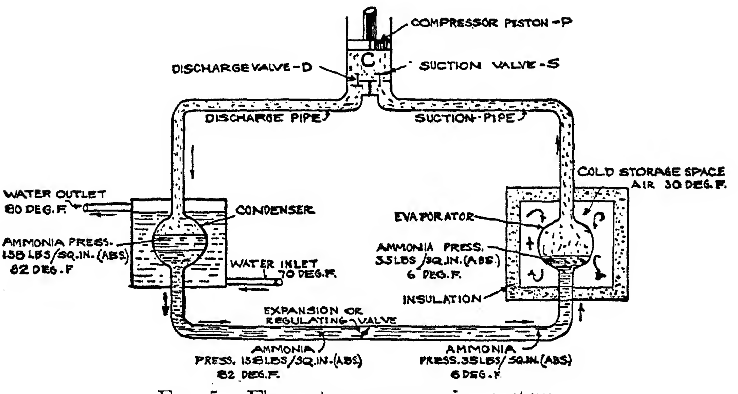


FIG. 5.—Elementary compression system.

purposes. Thus, if the temperature in the refrigerator is to be 25° F., the temperature of the vapor in the evaporator must be a few degrees lower in order to cause heat to be removed from the refrigerator by the evaporation of liquid refrigerant. The description of this compression system so far has explained only the low-pressure part of the process, that is, the right-hand side of the apparatus as shown in the figure. On the left-hand side is the *condenser,* which receives the compressed vapor of the refrigerant at high pressure. The temperature of the refrigerant in the condenser must be a few degrees above the temperature of the cooling water circulating through the condenser, in order that heat may pass from the vapor in the condenser into the circulation water used for cooling. The effect of this cooling by water is to condense the vapor of the refrigerant. After

the vapor has been condensed or, in other words, has become liquid, it passes through a regulating valve, generally called an *expansion valve,* which is really a reducing or throttling valve intended to reduce the pressure of the liquid refrigerant from the *high pressure in the condenser* to the *lower pressure in the evaporator.* After the refrigerant has passed through the expansion or regulating valve, where its pressure is reduced, and through the evaporator, it flows again into the suction pipe of the compressor.

Briefly, in the compression system, refrigeration is produced by the repeated process of compression, condensation, expansion, and evaporation.

The refrigerants which are most suitable for use in this system are "*manufactured*" vapors or gases. It would be too expensive and otherwise objectionable to discharge into the atmosphere the vapor or gas of the refrigerant after it has been used. For this reason, the refrigerant is used over and over again. If there are no appreciable leaks in the system, there will be very little loss of the refrigerant in long periods of time.

In the compression system, the action of the refrigerant in transferring heat from a *low* temperature to a higher temperature and then discarding the heat at the *higher* temperature may be compared to the action of a sponge which is used to lift water from a bucket. First, the sponge is compressed by the hand; it is then immersed in the water in the bucket. As the pressure of the hand is released, the sponge expands and absorbs water which may then be lifted out of the bucket with the former. If the sponge is now compressed somewhere other than over the bucket, the water in the sponge may be discarded, and when the sponge is again compressed and allowed to expand in the bucket it will absorb water as before. The repetition of this process with the sponge is like the repetition of events in the compression system of refrigeration.

The necessary parts of a compression system—the compressor, the condenser, the expansion valve, the liquid receiver, and the cooling coils of the evaporator—are shown in an outline drawing in Fig. 6. Arrows show the direction of flow of the refrigerant through the system. The compressor C may be operated by any suitable source of power A. Compressors may be operated by direct connection to electric motors, oil engines, or steam engines, or they may be driven indirectly by belts receiving their

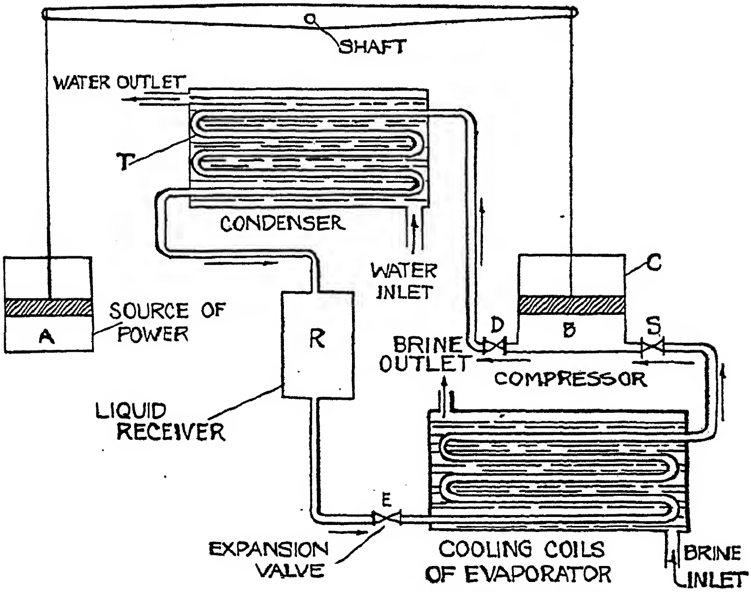


FIG. 6.—Outline of compression refrigerating system.

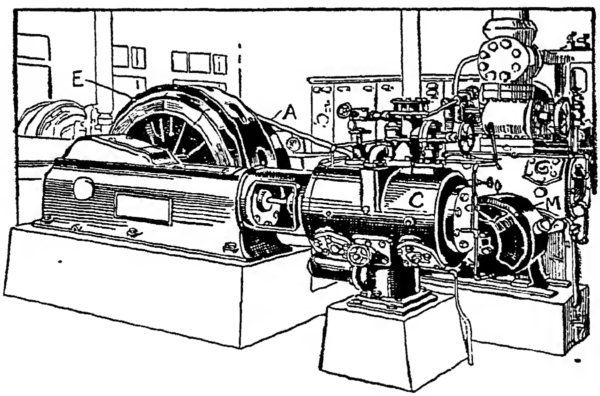


FIG. 7.—Modern compression refrigerating plant.

ɔower from electric motors. As shown in Fig. 7, the electric ɑotor E drives the compressor C.[1]

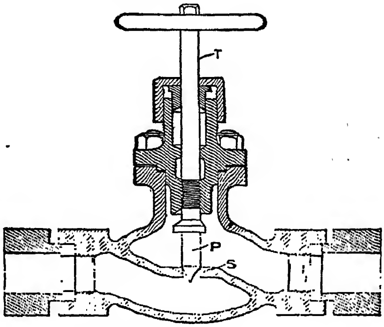
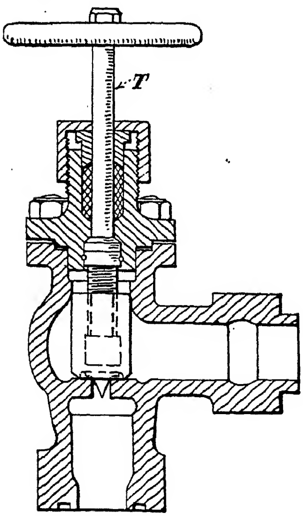


FIG. 8.—Needle-pointed expansion valve with screwed ends.

In the operation of the compressor in Fig. 6, the low-pressure vapor of the refrigerant is taken from the cooling coils of the evaporator in which the liquid refrigerant has previously been evaporated, and this vapor is compressed to a higher pressure in order to raise the temperature at which it condenses. After compression, the vapor of the refrigerant is discharged at a high pressure through the discharge valve D into the condenser. In the apparatus shown in the figure, the condenser consists of a coil of pipe submerged in a tank T of running water for cooling and condensing the vapor of the refrigerant. For the proper operation of the condenser, the temperature of the water for cooling must be lower than the temperature of the vapor of the refrigerant in any of the coils of the condenser. The water will then remove heat, and the vapor of the refrigerant will be changed to a liquid.



G. 9.—Angle type of needle-pointed expansion valve.

Liquid Receiver.—From the condenser, the liquid refrigerant flows into the liquid receiver R

[1] The heavy flywheel effect of the electric motor E is needed to make the rotary drive of the electric motor adaptable to the reciprocating motion of the compressor. The motor M is used only for starting. When the compressor is started, the synchronous motor E is used to drive it.

(Fig. 6), where the pressure is nearly the same as in the condenser, although its temperature may be somewhat lower than that of the vapor which enters the condenser. The liquid receiver is a storage space for the liquid refrigerant which would accumulate otherwise in the condenser. By the application of this receiver, there is no accumulation of liquid refrigerant in the condenser to reduce the effectiveness of its cooling surface.

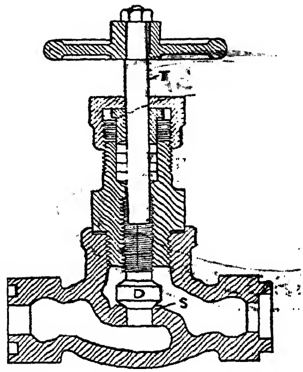


FIG. 10.—Expansion valve with flanged ends.

Expansion Valves.—The refrigerant passes from the liquid receiver into the expansion valve at E. This valve consists of a "needle-pointed" stem which makes possible a sensitive adjustment of the flow of the liquid refrigerant through the valve and into the coils of the evaporator. It is generally adjusted so that all the liquid refrigerant will be vaporized in the coils of the evaporator. *Needle-pointed expansion valves* are shown in Figs. 8 and 9; and a slightly different type, in Fig. 10.

Automatic Expansion Valve.—In an automatic refrigerating system, as used for example in household refrigerators, the expansion valve supplies the liquid refrigerant to the cooling coils of the evaporator as it is needed. The automatic valve of this kind, shown in Fig. 11, is moved by a spiral adjusting spring operating against the pressure of the vapor of the refrigerant in the evaporator. As the pressure in the evaporator is lowered, the diaphragm is pressed downward by the action of the spring and opens the needle-pointed valve to allow more liquid refrigerant to enter the evaporator. During the flow of the liquid refrigerant into the evaporator, the compressor, running at a constant speed, cannot take away all the vapor formed in the evaporator by the addition

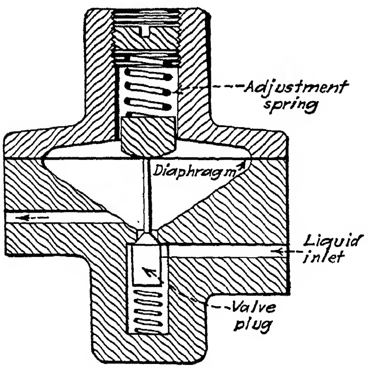


FIG. 11.—Automatic expansion valve.

of so large an amount of refrigerant, and there is an increase in pressure in the evaporator which forces the diaphragm upward and closes the valve. This action when repeated regularly supplies the liquid refrigerant to the evaporator as required. The valve is nothing more than a pressure-reducing valve and has one outstanding fault, namely, diaphragm failure. A suitable strainer must be located "ahead" of such a valve to prevent scale from getting under the seat of the valve.

Thermal Expansion Valve.—An expansion valve operated by the expansion of a fluid that is heated by the vapor of the refrigerant is shown in Fig. 12. In this valve there are two separate chambers, each filled with the vapor of the refrigerant. The expansion valve is regulated from a separate chamber in the pipe line carrying the vapor of the refrigerant away from the coils of the evaporator. The operation of this valve depends directly on the

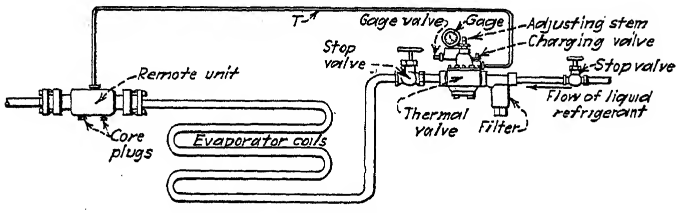


FIG. 12.—Thermal expansion valve.

temperature of the vapor of the refrigerant leaving the cooling coils of the evaporator. If the temperature of the vapor leaving the evaporator rises above a definite value, the pressure of the vapor rises correspondingly in the apparatus shown at the left-hand side of the figure called the "remote unit," and this pressure is transferred through the tube T to the thermal valve shown at the right-hand side, where this pressure is exerted on a suitable diaphragm to bend it downward against the resistance of a spring and the suction pressure. This movement of the diaphragm opens a needle-pointed expansion valve, which allows more refrigerant to enter the cooling coils of the evaporator. The admission of this additional supply of refrigerant reduces the temperature of the vapor leaving the evaporator and at the same time gradually decreases the pressure in the remote unit. This reduction in pressure in the remote unit affects similarly the pressure above the diaphragm in the thermal

valve which then closes and shuts off the flow of the refrigerant into the evaporator. By the operation of the remote unit and the thermal valve as thus explained, there is maintained a fairly constant pressure in the coils of the evaporator. In the operation of the thermal valve, as explained, it has the effect of making the vapor of the refrigerant slightly superheated (p. 65) when it leaves the coils of the evaporator.

This type of thermal expansion valve can be applied most successfully to automatic refrigerating plants that are operated by an electric motor.

In automatic electrical refrigerating systems that are started and stopped by a thermostat located in the refrigerated compartment, a magnetically operated valve is placed in the liquid line ahead of the thermal expansion valve. This magnetic solenoid valve is opened and closed electrically, according to whether the motor is running or stopped. This kind of protection is necessary as the thermal valve would otherwise completely fill the coils of the evaporator with liquid refrigerant, when the compressor stopped; and this liquid refrigerant would be drawn into the compressor with disastrous results when again started, as the cylinder of the compressor would be at least partly filled with an incompressible fluid.

The refrigerating effect of the system is produced in the cooling coils of the evaporator when the liquid refrigerant evaporates. As shown in Figs. 6 and 12, the coils of the evaporator consist of a coiled pipe which may be placed where the cooling effect is desired. The liquid refrigerant, after passing slowly through the expansion valve, enters the cooling coils of the evaporator in which a comparatively low pressure is maintained by the suction of the compressor. In becoming a vapor, the refrigerant absorbs heat from the surrounding substances in contact with the coils and thus cools them.

In places where electric current is obtainable at a reasonable cost, the compressor in modern refrigerating plants is driven by an electric motor. When electricity for power is not available, Diesel oil engines are frequently used as a source of cheap power. Formerly, steam engines were used almost exclusively for power in refrigerating plants. Figure 13 shows a compression refrigerating plant.

Refrigerant a Carrier of Heat.—In the compression system, the circulation of the refrigerant is from the compressor to the con-

denser, to the liquid receiver, through the expansion valve to the cooling coils of the evaporator, and then back again to the compressor. Thus, the refrigerant is actually a *carrier of heat.* The action of the compressor is similar to that of a pump, as it *lifts*

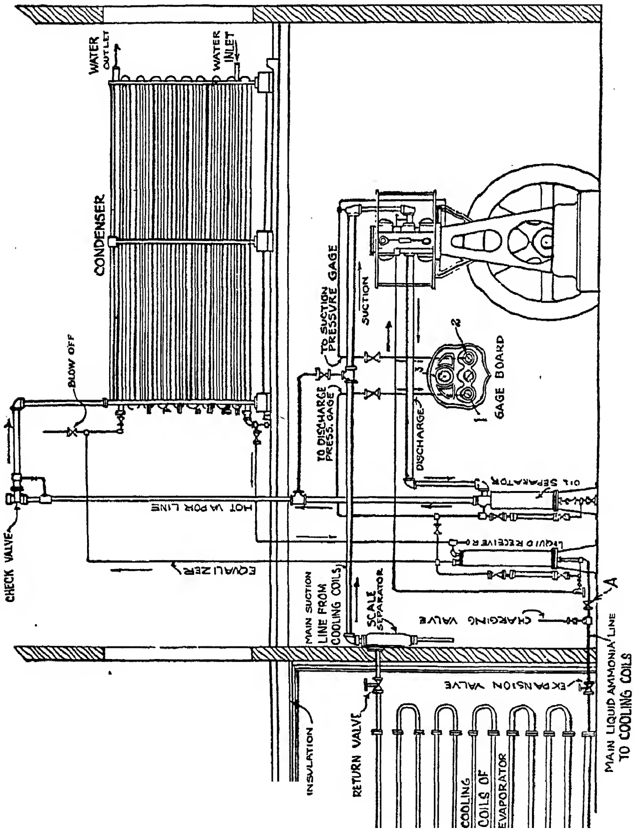


Fig. 13.—Complete compression refrigerating system.

heat at a low temperature from the cooling coils of the evaporator and delivers it to the condenser at a much higher temperature. When the vapor of the refrigerant is subjected to a high pressure by the mechanical action of the compressor, a certain amount of heat is added to the vapor, which raises its temperature. The

heat added in this way is carried along in the refrigerant to the condenser, where it is removed.

Ammonia Absorption System of Refrigeration.—Refrigeration by the absorption system differs only slightly from the compression system, the difference being that a coil supplied *alternately with steam and water* and fitted into a *closed pressure tank* filled with a mixture of refrigerant and water is used in place of a compressor. The evaporator, condenser, and the expansion or regulating valve are the same in the two systems. Briefly, the difference in the two systems is in the method of increasing the pressure between the evaporator and the condenser. In the compression system, the increase of pressure is brought about by mechanical means, that is, by the use of a compressor. In the absorption system, the increase in pressure is produced by heat supplied by means of steam which circulates through a coil of pipe.

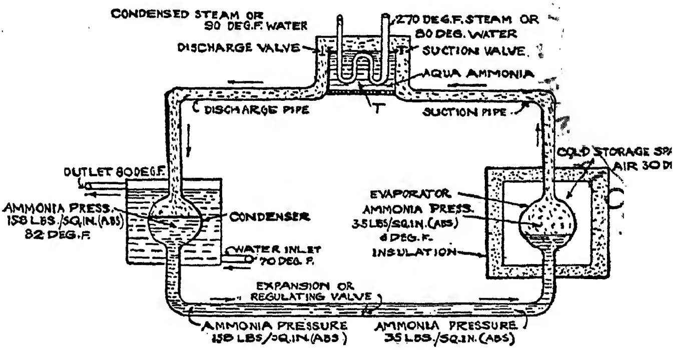


Fig. 14.—Elementary absorption refrigerating system.

At some temperatures, water has the property of absorbing many times its volume of ammonia vapor. For example, when water is at the temperature of 55° F., it will absorb about one thousand times its volume of ammonia vapor, but if the temperature of an ammonia solution is raised to, say, 80° F., ammonia vapor will escape freely from the liquid solution.

The absorption system of refrigeration is based on the principle of the *absorption of ammonia vapor by water at relatively low temperatures and the giving up of ammonia vapor when the mixture is heated.* Ammonia is the most suitable refrigerant for use in absorption systems. Mixtures of ammonia and water are called *aqua ammonia.*

In the operation of the absorption system in its simplest form, as shown in Fig. 14, the liquid ammonia which comes from the condenser flows through a short length of piping to the expansion or regulating valve, where its pressure is reduced in the same way as in the compression system. After expansion, the refrigerant passes on through other piping to the evaporator. From the evaporator, the low-pressure ammonia vapor passes upward into a *closed* pressure tank *T*, entering through the inlet or *suction valve.* The low-pressure ammonia vapor is absorbed by the "weak" *aqua ammonia* already in the tank. The absorption of ammonia vapor is accelerated by the method of cooling the aqua ammonia by passing cold water through the coil shown in the tank *T*.

When water is circulated through the coil in this tank, the ammonia gives up heat to the water, which is heated, for example, from 75 to 90° F. When the aqua ammonia in the pressure tank has absorbed all the ammonia which it can hold, the valve on the water supply is shut off, and the valve on the steam supply is opened so that steam can pass through the coil. In the apparatus in the figure, the temperature of the steam is 270° F. By giving up heat, the steam raises the temperature of the aqua ammonia in this tank from about 150 to 250° F. At the higher temperature, the aqua ammonia is reduced as the result of this rapid evaporation to a concentration (p. 235) of about 25 per cent of ammonia by weight. The steam supply is then shut off, and water is again passed through the coil.

The ammonia which boils off the surface of the aqua ammonia passes out through the discharge valve and the pipe leading to the condenser, where it becomes liquid by being cooled with the water which circulates through the condenser. This set of operations is repeated over and over again with the continuous circulation of the same supply of ammonia which is successively *evaporated, absorbed, distilled, liquefied,* and *expanded.*

In order to make more vivid the similarity of the absorption system to the compression system, the closed pressure tank *may be regarded as a compressor operated by heat* rather than by mechanical means. The absorption period in the pressure tank corresponds to the suction stroke of the compressor, while the period of increasing pressure in this tank takes the place of the compression stroke.

The closed pressure tank *T*, in Fig. 14, performs a double duty; in the first place, it absorbs the low-pressure ammonia

vapor, and by this absorption its concentration of ammonia is increased. The tank T serves also as a pressure generator when, by the application of heat from the steam coil, high-pressure ammonia vapor is driven off. In a more practical device for

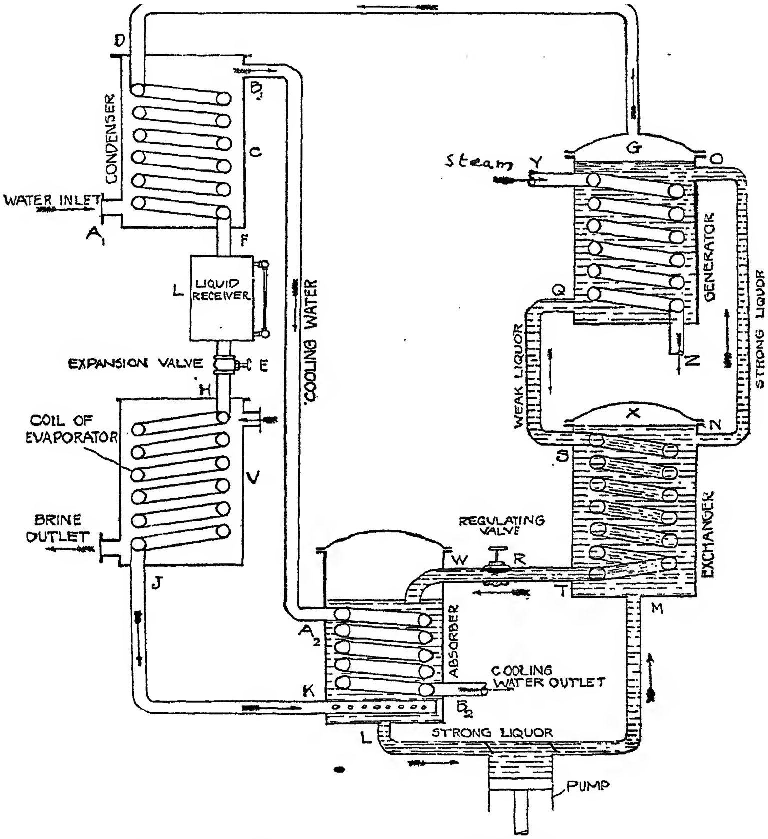


FIG. 15.—Outline of absorption refrigerating system.

the absorption system, there is one vessel, called the *absorber*, for absorbing the ammonia vapor, and another, called the *generator*, for increasing the temperature to the boiling point and driving off ammonia vapor at a high pressure. The absorber and generator will be described in detail in the following paragraphs.

When the amount of ammonia in an aqua-ammonia solution is relatively small, as, for example, after considerable vapor has been driven off, the aqua ammonia is called *weak liquor*. When, on the other hand, the concentration of ammonia is large, it is called *strong liquor*.

A *diagrammatic drawing* of a practical absorption system of refrigeration is shown in Fig. 15. The condenser C consists simply of a coil, submerged in water. The cooling water entering the condenser at A_1 and leaving at B_1 serves to condense the ammonia vapor which comes to the condenser from the *generator*, entering the condenser at D. The liquid ammonia formed by condensation is drawn off at F into the liquid receiver L, from which it flows through the expansion valve E into the coils of the *evaporator*. The condenser, expansion valve, and evaporator are exactly like those of the compression system. All of the remaining parts of the absorption system are different.

The ammonia vapor leaves the coils of the evaporator at J and passes into the *absorber* through a perforated pipe K. The purpose of this perforated pipe is to cause the ammonia vapor to rise through the weak ammonia liquor in the form of bubbles. The *comparatively cool* weak liquor absorbs the ammonia vapor so that it becomes strong liquor. The strong liquor is continuously removed from the bottom of the absorber by the pump P, which forces it into the exchanger X, entering at M, leaving at N, and then flowing back into the *generator* at O.

The strong liquor entering at the top of the generator is heated by the steam coils YZ, the steam entering at Y. In these coils, the steam condenses, and the condensed steam is drawn off at Z. In condensing, the steam gives up heat to the strong liquor, thus raising its temperature. This increase in temperature drives off ammonia vapor from the strong liquor.

Strong liquor is lighter than weak liquor, or, in other words, the specific gravity of strong liquor is less than the specific gravity of weak liquor. The weak liquor formed in the generator is drawn away at the bottom and flows out at Q into the exchanger at S. The weak liquor flows out of the exchanger at T through a regulating valve R into the upper part of the absorber at W.

The purpose of the regulating valve R is to reduce the pressure of the weak liquor. The weak liquor leaving the exchanger is at the high pressure in the generator, and its pressure is reduced by this valve to the lower pressure in the absorber. The pressure

in the absorber is about the same as that of the *ammonia* entering the absorber from the coil of the evaporator.

Although not essential to the operation of the absorption system, the *exchanger* is a heat-saving device and is useful in reducing the operating costs. The strong liquor leaving the absorber is comparatively cool and is heated later in the generator. On the other hand, the weak liquor *leaving the generator* is at a high temperature. This liquor must be cooled either before reaching the absorber or in the absorber. The purpose of the exchanger is to *transfer heat* from the hot weak liquor coming from the generator to the cool strong liquor going back to the generator. In doing this, the weak liquor is somewhat cooled, while the strong liquor going to the generator has its temperature raised. By the use of this device, there is then a saving in the amount of cooling water required by the absorber. There is also a saving in the amount of steam required to raise the temperature of the strong liquor in the generator.

Although there is a saving of a large amount of heat by the use of the exchanger, there is only an "equalling" of temperature. This means that the heat in the weak liquor cannot raise the temperature of the strong liquor to that of the generator. On the other hand, the strong liquor cannot cool the weak liquor to the temperature of the weak liquor in the absorber. Because of this, some heat must then be added to the strong liquor in the generator, and some heat must be removed from the weak liquor in the absorber.

The weak liquor, in passing through the exchanger, is cooled a few degrees more than the temperature of the strong liquor is raised. This is because the weight of strong liquor passing through the exchanger in a given time is greater than the weight of the weak liquor passing through the exchanger in the same time. For example, if the weight of strong liquor entering the generator is 10 pounds and the weight of the weak liquid returning to the absorber is 9 pounds, the 9 pounds of weak liquor will be cooled through a greater range of temperature than the number of degrees the 10 pounds of strong liquor will be heated.

The exchanger consists of either a vertical or a horizontal steel drum, capable of carrying the generator pressure. This drum contains a coil of pipe. The weak liquor flows from the generator through the coil of pipe to the absorber. The strong liquor

surrounds the coil and is pumped through the exchanger X into the generator.

The weak liquor flowing out of the exchanger will have a higher temperature than that of the absorber. This means that there is considerable heat in the weak liquor when it enters the absorber; besides this heat, there is also the heat generated by the absorption of the ammonia vapor by the liquor in the absorber. In order that the liquor in the absorber may absorb ammonia vapor, the heat brought in by the weak liquor and the heat generated by absorption must be removed. Because of this, it is necessary to cool the absorber. This can be accomplished by circulating the cooling water, which is discharged from the condenser at B_1, through the submerged coil of the absorber. The temperature of cooling water leaving the condenser at B_1 will be about 85 or 90° F. This temperature is sufficiently low for cooling the liquor in the absorber to a temperature that will give the desired absorption. Sometimes the weak liquor is cooled by a separate cooler which is entirely separate from the absorber. This separate cooler receives a part or all of the cooling water from the condenser, and in that case the absorber is cooled by an independent water supply.

The amount of liquor which must be circulated to absorb one pound of liquid ammonia containing no water (*anhydrous ammonia*) depends on the strengths of the strong and the weak liquor. Because of this, it is necessary to have the strong liquor entering the generator as strong as possible and the weak liquor going to the absorber as weak as possible. When this is accomplished, a comparatively large amount of ammonia vapor can be driven out of the liquor in the generator. It is necessary then only to circulate a comparatively small amount of liquor, to produce a given refrigerating effect. A high temperature in the generator means a large amount of ammonia vapor driven off, while the liquor returning to the absorber will be very weak. On the other hand, a low temperature in the absorber will result in a large quantity of ammonia vapor being condensed and absorbed, and the strong liquor going to the generator will have a large percentage of ammonia. It is essential, then, to have a high temperature in the generator and also to have a large amount of cooling in the absorber.

An actual layout of an absorption refrigerating system is shown in Fig. 16. Taking the condenser as the starting point

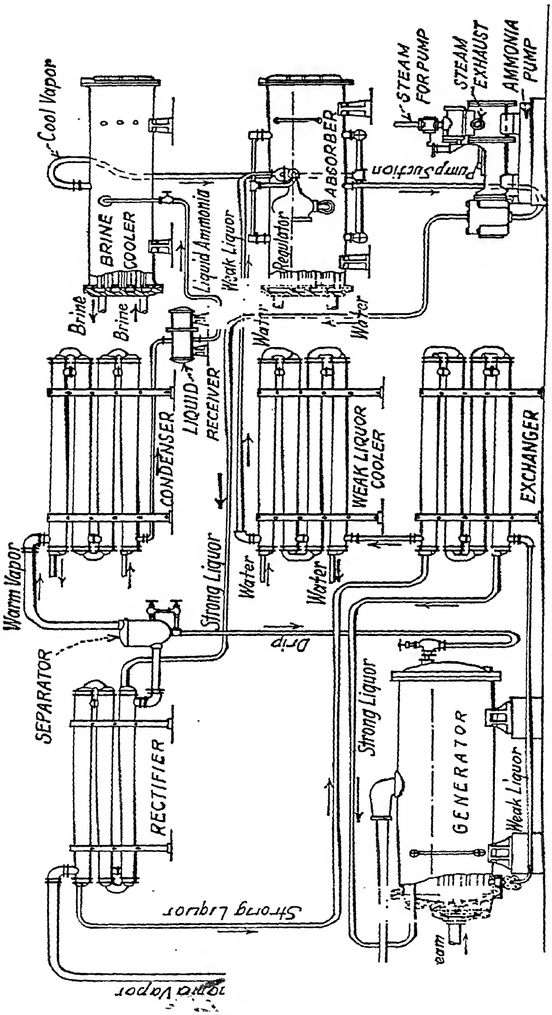


FIG. 16.—Tubular absorption system.

in this description, the liquid ammonia flows from the bottom of the condenser, by gravity, first into the liquid receiver and then through an expansion valve (not shown) into the coil of the evaporator, which is located, in this case, in a brine cooler. Then, after absorbing heat by cooling the brine during its evaporation, the ammonia passes as a cool vapor to the bottom of the absorber.

In the absorber, the ammonia vapor coming from the evaporator coil is absorbed by the liquor already in it and gives up heat to the cooling water. The strong liquor resulting from the absorption of the ammonia vapor is then transferred from the absorber by the ammonia pump.

The absorption system is usually provided with a *rectifier* for the purpose of thoroughly drying the ammonia vapor before it enters the condenser. This is a device which is used to condense the water vapor from the mixture discharged by the generator. This condensed water vapor, of course, absorbs some ammonia, making a strong liquor which must be returned to the generator to be used over again. As shown in the figure, this liquid is removed from the piping system at the bottom of the separator and is carried to the generator through the "drip" pipe.

After passing through the rectifier, the strong liquor enters the top of the exchanger, and, after passing downward through this apparatus, it is discharged into the generator, near the top.

In the generator, steam is used to heat the ammonia liquor in order to distill the ammonia vapor and steam, which pass out and upward to the rectifier from a connection shown at the top of the generator. As the result of removing ammonia vapor, weak liquor flows from the bottom of the generator, and passes because of pressure difference into the exchanger. After passing upward through the exchanger, the weak liquor leaves at the top and then enters the bottom of the weak-liquor cooler. Cooling water circulates through this apparatus by entering at the inlet at the top and leaving at the outlet at the bottom. This cooling water reduces the temperature of the weak liquor so that when it reaches the absorber it will be at the best temperature for the most efficient operation. From the weak-liquor cooler, the weak liquor discharges through a liquor regulating valve into the absorber, where it reabsorbs ammonia vapor which comes into the absorber from the coil of the evaporator.

In the meantime, the *ammonia vapor and steam,* which discharged upward from the top of the generator, have gone to the

top of the rectifier, where the strong liquor from the ammonia pump flows in countercurrent direction to the direction of flow of

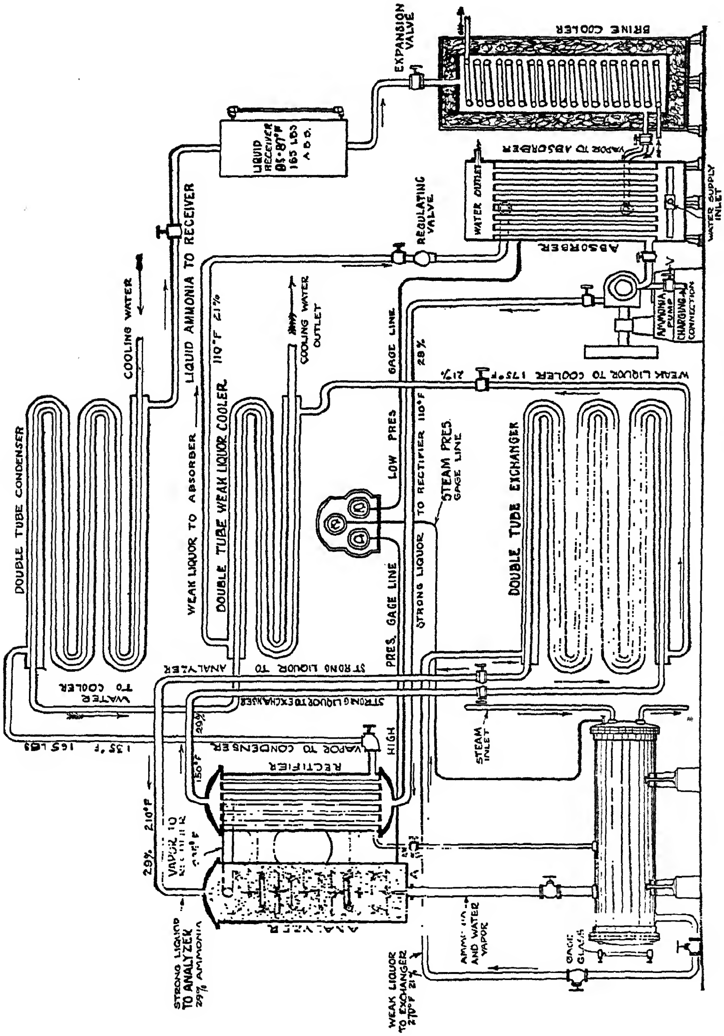


FIG. 17.—Complete absorption refrigerating system.

the ammonia vapor and steam, and the strong liquor takes up in the rectifier some heat from the ammonia vapor and steam,

thereby condensing the steam. The rectifier has a use here somewhat similar to that of a regular steam condenser. The moisture resulting from this condensation collects in the separator shown at the right of the rectifier. The drain or drip-pipe of the separator carries the moisture back to the right-hand end of the generator.

The warm ammonia vapor passes from the bottom of the rectifier through the separator into the top of the condenser. Here the heat taken up from the ammonia vapor by the cooling water condenses the ammonia vapor. The ammonia in the liquid state then passes on again through the expansion valve

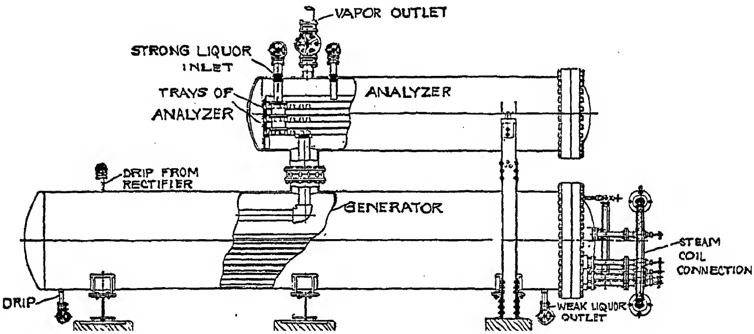


FIG. 18.—Typical generator and analyzer for absorption system.

and the coil of the evaporator in the brine cooler, thus completing its circuit.

Absorption refrigerating systems will vary somewhat in the arrangement and methods of connecting the various parts. In some plants, a "drying" apparatus, called an *analyzer*, is recommended to be inserted in the system between the generator and the rectifier. The analyzer is used as a companion device to the rectifier to bring about a large transfer of heat between the hot vapors coming from the generator and the strong liquor which circulates through these two devices.

A typical layout of an absorption system in which an analyzer is used to supplement the rectifier is shown in Fig. 17. As shown here, the analyzer consists of a vertical cylinder, containing a stack of shallow pans, one placed above the other. A different arrangement of the analyzer with respect to the generator is shown in Fig. 18. The different cycles or paths of the liquids and

vapors are shown by arrows in Fig. 17.
now is that of the strong liquor from the
ammonia pump, and into, first, the exchai
analyzer. The discharge of liquor from the ana.
ward by gravity into the generator. Another cycle of importance as showing the use of the analyzer is that of the anhydrous ammonia, which, after leaving the generator at A, passes first, through the "rain" of strong liquor in the analyzer and then through the rectifier on its way to the condenser. From the condenser, the liquid refrigerant flows through the liquid receiver and the expansion valve into the coil of the evaporator, where the liquid ammonia again becomes a vapor. This ammonia vapor then passes into the absorber, where it is absorbed by the weak liquor, and is pumped as strong liquor through the rectifier, exchanger, and analyzer. From the analyzer, the strong liquor falls by gravity back into the generator. There is also the cycle of the brine, which passes through the brine cooler, where it gives up heat to the cooling coil of the evaporator and then passes through the refrigerating coils, where it takes up heat on its way back to the brine cooler.

Analyzers and Rectifiers.—In driving off the ammonia vapor in the generator, some water will also be driven off; hence, a mixture of ammonia and water vapor will result. If this mixture is permitted to pass over into the condenser (where it will be condensed) and then passes on into the liquid receiver and into the coil of the evaporator, the liquid ammonia in the evaporator coil will, of course, evaporate, leaving the water, which will eventually fill the coil. Such an accumulation of water in the evaporator coil will obviously lower its efficiency. On the other hand, if the water vapor passes along with the ammonia vapor and enters the condenser, settling in the bottom, the water will accumulate and also reduce its efficiency. It is very necessary, therefore, to remove this water vapor from the ammonia vapor, and this is accomplished very successfully by the use of the *analyzer* and *rectifier* in combination.

The purpose of a rectifier when used in conjunction with an analyzer is to remove about 7 per cent of water vapor, which remains mixed with the ammonia vapor, thus bringing only pure anhydrous ammonia vapor to the condenser. The rectifier shown in Fig. 17 consists of a cylindrical drum containing tubes. The

strong liquor passes through them. The strong liquor leaving the absorber is quite cool, as compared to the hot ammonia and water vapors, so that it will absorb heat from the hot vapors. In doing so, the water vapor is cooled and is condensed on the surfaces of the tubes. This water absorbs ammonia in the rectifier and the mixture is then drained through a drip pipe into the generator. The ammonia vapor now freed from water vapor passes on into the condenser.

Present Use of Analyzers.—Some manufacturers recommend the use of the analyzer, while some do not. Analyzers were used frequently in the past, but, at present, they are to be used only under special conditions.

Field of Application of Absorption System.—The absorption refrigerating system has its own particular field of application. It operates quite economically at low evaporator pressures. At evaporator pressures below 8 to 10 pounds suction-gage pressure, the ammonia absorption system will show more economical results, in most cases, than the ammonia compression system when operated by electric motors or compound condensing steam engines.

The application of the compound (two-stage) ammonia compression system (p. 57) is restricting somewhat the further application of the absorption system. *In plants, however, where there is available a quantity of low-pressure steam, it is advantageous, in some cases, to install the absorption system.* As previously stated, *when very low temperatures are needed,* the absorption system will produce these low temperatures very economically.

The inefficiency of the absorption system results from the fact that the amount of heat which is contributed by the steam in the pressure tank and then removed by the cooling water in the condenser is very much larger than the heat which is absorbed by the expansion and boiling of the cool liquid in the evaporator. This fact accounts for the relatively low thermodynamic[1] efficiency of the absorption system when compared with the compression system. This system uses more cooling water than a compression system.[2]

[1] The thermodynamic efficiency of the absorption system of refrigeration is explained on pp. 259 and 294.

[2] The absorption systems of refrigeration are often equipped with electric heating devices which require, when operating, from 1 to 2 kilowatts and are in use for only an hour or two a day. This type of machine if installed

The cost of an ammonia absorption refrigerating plant is about 65 per cent more than the cost of an ammonia compression plant, exclusive of piping, insulation, and buildings.

The experts of the National Electric Light Association state that, until some absorbent is found which will not heat so much as water in absorbing the ammonia vapor, there is little hope for the commercial success of the ammonia absorption system in small plants which must be *heated by electricity.*

Silica Gel or Adsorption System.—In the *ad*sorption system of refrigeration a substance known as *silica gel,* which is a hard glassy material resembling clear quartz sand of the chemical formula SiO_2, can be used advantageously. Silica gel is extremely porous, but the pores are so small that they cannot be seen with a microscope. It has been found that the voids constitute 41 per cent of its volume. The presence of these voids gives silica gel its ability to adsorb relatively large quantities of vapor. In fact, silica gel when placed above water in a closed vessel will adsorb water vapor to the extent of 25 per cent of its own weight. Now, if the silica gel which has adsorbed water vapor is heated, the water vapor will be driven off leaving it in a state ready again to adsorb vapor. This action is purely physical as there is no chemical reaction upon the silica gel.

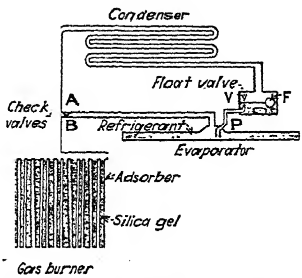


FIG. 19.—Silica-gel refrigerating system.

These principles have been used in producing the refrigerating system shown in Fig. 19. The apparatus consists of three main parts, namely, (1) the *ad*sorber (containing the silica gel), (2) the evaporator, and (3) the condenser. The *ad*sorption system resembles the compression system, with the compressor replaced by the *ad*sorber. The *ad*sorption of the refrigerant by the silica gel corresponds to the suction stroke of the compressor and the "activating" of the silica gel to the discharge stroke.

The operating cycle may be shown by referring to Fig. 19. Assuming that the silica gel in the *ad*sorber has been activated,

in large numbers is an undesirable load for an electric lighting system. It would be a desirable improvement to reduce the kilowatt capacity of heat elements to, say, 500 watts, so that the maximum load on the electric lighting . would not be so large.

it will *ad*sorb the vapor of the refrigerant from the evaporator, thus lowering its temperature. The escape of any of the vapor of the refrigerant from the condenser is prevented by the float valve V and the check valves A, shown in the vertical pipe C.

When the silica gel in the adsorber has become saturated with the refrigerant it is heated by means of a suitable burner as shown. Heating the silica gel drives off the refrigerant which being under pressure rises in the pipe C to the condenser where it liquefies. The liquid then returns to the evaporator through the float valve. The check valve B (p. 64) in the horizontal pipe prevents the vapor of the refrigerant from entering the evaporator. When activation of the silica gel has been completed the source of heat below the adsorber is removed, and as soon as the silica gel has cooled sufficiently the adsorption phase

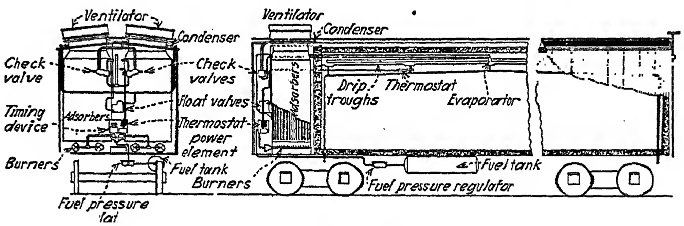


FIG. 20.—Silica-gel refrigerator car.

begins automatically. In actual operation the heating period is shorter than the adsorption period, and by dividing the adsorber into two sections and heating these sections alternately, continuous refrigeration is produced. Sulphur dioxide is the refrigerant generally used. Figure 20 shows how this system is applied to refrigerator cars.

Vacuum System of Refrigeration.—Water is sometimes used as a refrigerant in refrigerating systems by causing it to boil by merely reducing the pressure in the container with a vacuum pump. In this case, the refrigerant (water) is liquid at ordinary temperatures and pressures and is always conveniently obtained. In this respect, it has important advantages over the so-called *gas* refrigerants. On the other hand, when a very low pressure is obtained in a container, there is likely to be trouble from air leaks.

Vacuum systems using water for the refrigerant have not been used very much, but, recently, an apparatus of this type has been developed for use in air-conditioning work. A centrifugal type of high-vacuum pump is preferably used, and considerable

capacity is obtained in this type in small dimensions; likewise, very low pressures (vacuum of 29 inches of mercury or about ½ pound per square inch absolute) are obtained. In this process, the water evaporates quite freely at ordinary atmospheric temperatures and a vacuum of 29 inches of mercury and is condensed on the high-pressure side of the system at a vacuum of about 18 inches. It will be noted, therefore, that all pressures throughout the system are less than atmospheric and that none of the refrigerant can, therefore, leak out. The air which leaks into the system is, from time to time, sucked out by the operation of a very efficient air ejector. The principal advantage of this system is that it can be installed in much smaller space than would be required for an ammonia compression system. The experts of the National Electric Light Association make the following comment as to the possible future application of this type of machine: "The vacuum machine gives some promise because with proper machine design and the right refrigerant only moderate volumes of vapor need be pumped, and the machine may be small."

Vap-air System of Refrigeration.—A combination system of refrigeration has been worked out which is based partly on the vacuum system and partly on the air-machine system. In this combination system, a vacuum pump reduces the pressure in the container for the liquid refrigerant and, at the same time, compresses the residual air in the system and allows it to expand through the liquid. When the air is expanded or is being blown through the liquid, it gives up some energy, because some work is being done in greatly extending the surface. The extension of the surface of the liquid proportionately increases the amount of evaporation. The process is then further aided by making the evaporation much more effective as a result of reducing the total pressure at the surface of the liquid by rapidly removing the vapor and air mixture (vap-air) with a vacuum pump. Briefly, the vap-air system consists of two of the simpler systems combined into one. In the first place, compressed air expands against the resistance of a body of liquid, and in this expansion it becomes cold. In the other part of the system, a violently boiling liquid causes heat to be absorbed as latent heat.

In further explanation of this system, it may be stated that it is not subject to some of the limitations of the air machine, in

that an expansion engine or motor is not required. As regards the vacuum machine itself, furthermore, not so high a vacuum is necessary as with the vacuum system, so that there is not so much difficulty in preventing air leaks in the system. There is, also, the further advantage that the pressure throughout the system is about atmospheric. The pressure in the evaporator is the sum of the vapor pressure of the refrigerant and the air pressure,[1] but the temperature is limited only by the pressure of the vapor.

There are several fundamentally new thermodynamic principles applied in the vap-air system, and in this respect it is of more ordinary interest. In the commercial development of the device, there is commendable conservatism, so as to avoid, if possible, the disastrous results of premature commercialism.

Absorbers.—The various kinds of absorbers can be classified in three groups: (1) the wet absorber, (2) the dry absorber, and (3) the wet-and-dry absorber. The absorbers shown in Figs. 15 and 17 are both of the wet type. In the wet absorber, the drum is nearly full of liquor. The ammonia vapor enters at the bottom of the absorber and passes up through the liquor, by which the vapor is absorbed before it reaches the surface.

It is necessary to cool the absorbers, and, for this purpose, coils of pipe or straight tubes are provided to carry the cooling water. If the cooling is done by coils of pipe, there are generally several concentric coils. This permits a large cooling surface in a small space. The ends of these coils are each connected to common headers. Owing to the fact that straight tubes are easy to clean, they are more often used than coils.

The *dry absorber* shown in Fig. 21 is somewhat different from a wet absorber. In this absorber, the ammonia vapor enters at the left of the top below the perforated plate, and the weak liquor enters at the top of this plate. The weak liquor then passes through the perforated plate, which causes it to fall in the form of rain. This perforated plate also distributes the weak liquor evenly over the entire cross-section of the drum. The mixing of the ammonia vapor and the weak liquor in this manner causes the vapor to be quickly absorbed, as there is a large liquor surface exposed to the ammonia vapor. The strong liquor

[1] an explanation of combined vapor pressures (Dalton's law), see p. 191 and Moyer, Calderwood, and Potter, "Elements of Engineering Thermodynamics," 4th Ed.

thus formed is collected at the bottom of the drum and is removed by the liquor pump. The cooling water enters at the bottom, passes through the coils, and leaves near the top. This arrangement makes the absorber work efficiently, as it operates on the counter-current principle. Thus, this arrangement requires the smallest amount of cooling surface. In Fig. 21, only one coil is shown, although, generally, there are several. This type of absorber may also be made horizontal, but, as the floor space required is greater, the vertical type is preferred.

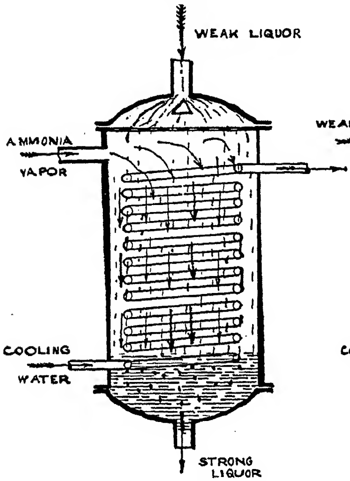


Fig. 21.—Dry absorber.

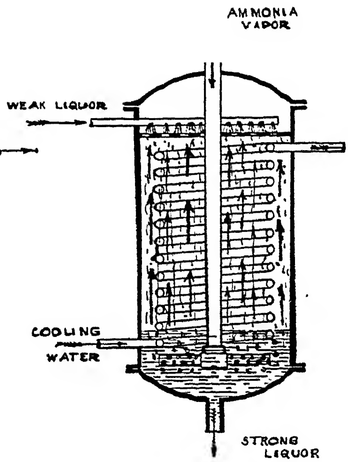


Fig. 22.—Wet-and-dry absorber.

The dry absorber is objectionable, as the cold ammonia vapor on entering may come into direct contact with the cooling-water tubes. This occurrence is likely to freeze the water in the tubes. It can be prevented, to some extent, by making the dry absorber so as to enter the vapor in such a way that it does not come immediately into direct contact with the cooling tubes. The vapor will then be warmed before it comes itself into contact with the tubes or coils.

A *wet absorber* is filled with liquor, and the ammonia vapor passes upward in the form of bubbles. A *dry absorber* contains only a small quantity of liquor, and the vapor is absorbed by contact with the rain of liquor.

An absorber utilizing both of these systems is called a *wet-and-dry absorber* (see Fig. 22). The wet-and-dry absorber contains

a small amount of liquor. The ammonia vapor enters at the top and passes down through a vertical pipe. This pipe extends to the bottom where it is connected to a perforated pipe. The ammonia vapor leaves this perforated pipe and passes up through the small quantity of liquor. That which is not absorbed by the liquor then rises and meets the rain of weak liquor which absorbs it. The weak liquor enters at the top and is sprayed upon a perforated plate, which causes it to fall over the cooling coils. This arrangement has the advantage that the ammonia vapor is effectively heated before coming into contact with the cooling coils. This eliminates the possibility of freezing the cooling water.

When using dry or wet-and-dry absorbers, the weak liquor is effectively cooled by the cooling coils, thus doing away with the weak-liquor cooler, saving the cost of a piece of apparatus, and reducing the expense for repairs.

In order to obtain the best results, the pressure in the absorber and in the coil of the evaporator should be nearly the same. By carrying a temperature as high as possible in the coil of the evaporator, there will be the smallest drop in temperature between the ammonia vapor in the coil of the evaporator and the brine, this condition being necessary for the best efficiency (p. 256). As stated before, the liquor leaving the absorber should be as strong as possible, so that it will be necessary to raise its temperature through only a small range in the generator to obtain adequate vaporization. If, however, the temperature of the strong liquor is too high, the liquor pump will race and slip, as the vapor will be formed in the pump. This action is similar to that of a boiler feed pump when it handles very hot water.

Rectifiers.—The rectifier shown in Fig. 17 is cooled by the strong liquor pumped directly from the absorber. As this liquor is comparatively cool, it is relied upon to condense all the water vapor which may enter the rectifier with the ammonia vapor. Such an arrangement is somewhat objectionable, as the cooling action which takes place is nearly constant and cannot be easily controlled or adjusted. This cooling action is dependent on the quantity and temperature of the strong liquor pumped through the rectifier. It may happen that the quantity and temperature may not be just right to condense all the water vapor suspended in the ammonia vapor. If these conditions cannot be adjusted to condense all of the water vapor, part will pass into the con-

denser, where it will condense. If this occurs, watery ammonia will be supplied to the evaporating coils. On the other hand, the quantity and temperature of the strong liquor can be such that all of the water will be condensed, and if the quantity and temperature of the strong liquor cannot be adjusted, some of the ammonia vapor will also be condensed. When this occurs, the liquid ammonia formed is returned by the drip pipes to the generator. This liquid ammonia produces no refrigerating effect, and its evaporation in the generator and circulation in the system are a loss. The rectifier in Fig. 17 must be designed to meet the required operating conditions in order to work satisfactorily. If this is done, the rectifier will give satisfaction, but it has the disadvantage of not being flexible.

FIG. 23.—Double-tube rectifier.

In some absorption systems, the rectifier is cooled by the water discharged from the condenser.

If the rectifier is cooled by water, it is constructed similar to a condenser but has several taps for draining the liquor collecting at the bottom. Such rectifiers may be of the double-tube type or of the atmospheric type. Figure 23 shows a water-cooled *double-tube type of rectifier*. The method of draining off the liquor collected at the bottom is clearly shown. The only difference between this rectifier and the atmospheric rectifier is that the atmospheric rectifier has single tubes and water is sprayed on the top tubes and then falls over those below.

Because of the importance of the rectifier in the absorption system, it should receive particular attention and care. The economy and capacity of this system are controlled by the cooling substance. In case the cooling substance is water, it should be carefully regulated to give the proper cooling, following the conditions of cooling already stated. The refrigerating effect of all the ammonia which is returned by the drips to the generator is

wasted, as it simply evaporates over and over in the generator. This reduces the amount of liquid anhydrous ammonia available for producing refrigeration.

In order to have the rectifier work at its best, the vapor leaving the rectifier should have a temperature of 20 to 40° F. above the temperature corresponding to the pressure. A thermometer is generally placed between the rectifier and the condenser. By observing this thermometer, the temperature of the vapor entering the condenser may be obtained. If the condenser pressure gage is also observed, the operator can determine if the amount of cooling in the rectifier is properly regulated. To do this, find the temperature corresponding to the absolute pressure in the ammonia tables and take the difference between the thermometer reading and the temperature found in the tables. It is also well to test the drip liquor occasionally to see if it contains too much ammonia; it should be hot and contain as little as possible. This testing should be done without raising the temperature of the vapor leaving the rectifier to such a value that it will be more than 40° F. above the temperature corresponding to the pressure in the condenser.

Condensers.—A water-cooled condenser should cool the refrigerant to nearly the same temperature as the entering cooling water. If the condenser is to be very efficient, it should operate on the *counter-flow* principle. When this principle is used, the *entering* warm *vapor* of the refrigerant is cooled by the surfaces in contact with the water *leaving* the condenser, and the liquid refrigerant on *leaving* comes into contact with the surfaces cooled by the *cold entering* water. This permits the liquid refrigerant to be cooled within a few degrees of the temperature of the cold cooling water. On the other hand, if the compressed vapor and the cooling water travel through the condenser in the same direction, called *parallel flow,* the warm vapor first comes into contact with the coldest surfaces, and the leaving liquid refrigerant comes into contact with surfaces having the highest temperature. These temperature changes for counter-flow and parallel-flow condensers are shown diagrammatically in Fig. 24 for the conditions of the same initial temperatures for both the vapor of the refrigerant and of the cooling water; the diagrams being laid out in both cases for a rise in temperature of 10° F. of the cooling water in its passage through the condenser. The reduction in temperature in the counter-flow condenser is represented by the

vertical distance between a and b and in the parallel-flow condenser by the vertical distance between a and b', the point b for the counter-flow condenser being considerably lower than the point b' for the parallel-flow condenser. In parallel-flow operation of a condenser, cooling water on leaving the condenser is only about 10 to 20° F. warmer than on entering, and, obviously, the liquid refrigerant is not cooled to a temperature so low as it would be by the counter-current principle. This reduces the available amount of refrigeration for cooling purposes, because there still remains considerable heat in the liquid refrigerant. Another disadvantage of parallel flow in a condenser is the possibility of re-evaporating some of the refrigerant. This is likely to take place when the liquid refrigerant formed by contact with the cooler surfaces comes later into contact with surfaces which are at a higher temperature.

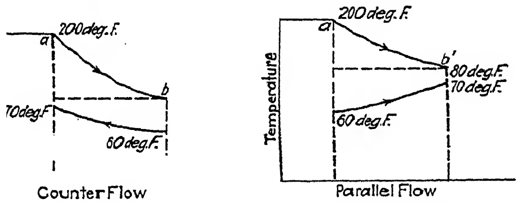


FIG. 24.—Temperatures in counter-flow condenser compared with parallel-flow condenser.

Classification of Condensers.—Condensers may be divided into four distinct classes: (1) the submerged condenser, (2) the atmospheric condenser, (3) the double-pipe condenser, (4) the shell-and-tube condenser, (5) the multipass condenser.

Submerged Condenser.—The simplest condenser for a refrigerating system is the submerged type. It consists of coils of pipe containing the compressed vapor of the refrigerant which are submerged in a tank of water. The ends of the coils are brought out at the top and at the bottom of the tank, thus avoiding submerged pipe joints. If these coils are made with submerged joints which are not tight, the escaping refrigerant would not easily be detected, because it would be absorbed by the water and pass off. The high-pressure vapor from the compressor enters at the top of the tank, and the liquid refrigerant is drawn off at the bottom. The cooling water enters the tank through a pipe connected at the bottom. The water gradually rises as

it is heated and passes off through an overflow pipe at the top. This condenser is operated on the *counter-flow* principle.

The submerged condenser has been found inefficient and is rapidly going out of use. The inefficiency is caused by the large amount of cooling water which passes through the condenser, much of which absorbs only a little heat. Furthermore, air bubbles collect on the coils of pipe and retard the transfer of heat. For these reasons, it is necessary to use 20 per cent more

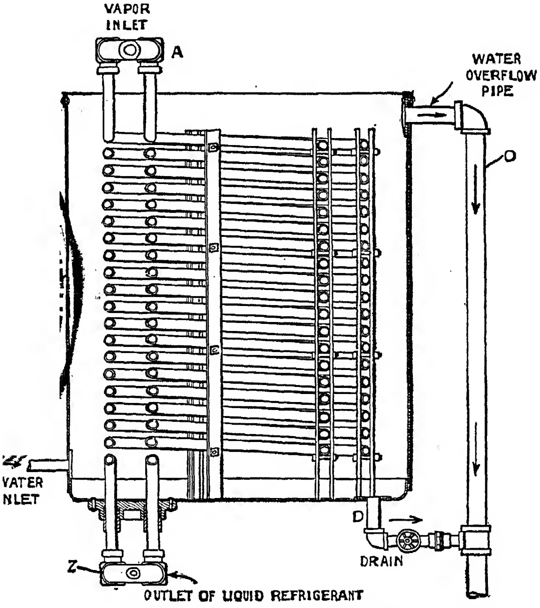


FIG. 25.—Submerged condenser.

circulating water in submerged condensers than in some other types.

Figure 25 shows a typical submerged condenser suitable for small plants or where the mist from falling water of atmospheric condensers would be objectionable. The vapor of the refrigerant enters at the top and flows downward, and the cooling water enters near the bottom of the tank T which surrounds the condenser coil and discharges into the overflow pipe O at the top. A drain pipe D at the bottom is to be used occasionally to remove all the water with accumulated sediment and scale.

Atmospheric Condenser.—The atmospheric condenser has found favor in recent years. This type of condenser consists of several vertical rows of horizontal pipes. Each length of pipe is joined to the next by return bends, and several lengths make up a so-called *stand*. When several rows or coils are used, they are connected into a common header. The number of rows varies with the capacity of the plant. In this form of condenser, the vapor of the refrigerant is in the coils, and the cooling water is allowed to flow over their outside surfaces. At the top of each coil is a trough with small holes. This trough distributes cooling water as "rain" upon the upper pipes of each coil, and the water which is not evaporated falls in streams over the surfaces of the lower pipes to be collected in the condenser pan and drawn off.

Cooling water distributed in this way increases the transfer of heat and removes heat from the refrigerant with a comparatively small amount of water. The evaporation of a small amount of water, in this way, absorbs a large quantity of heat, because the latent heat of evaporation of the water is large.

In atmospheric condensers, it is impossible to utilize fully the counter-current principle, because the cooling water must flow from the top to the bottom. The compressed vapor of the refrigerant enters at the bottom and in rising through the coils is condensed. The liquid formed then flows toward the bottom, where it comes into contact with warm surfaces and vapor. This will cause a portion of the liquid refrigerant to re-evaporate. This difficulty may be partly overcome by draining off the liquid through "bleeder" tubes which are tapped into the lower pipes of the coils.

The cooling effect produced by the evaporation of the water can be increased by locating the condenser on a roof, where it will receive a good circulation of air, thus removing the air which has become saturated with water vapor. If the condenser is to be placed on a roof, it should be shielded from the direct rays of the sun.

Some of the water spray may be blown away from the surfaces of the pipes by a strong wind. The cooling effect of atmospheric air blowing over the rack of pipes reduces the amount of cooling water required in cold weather.

A typical atmospheric condenser is shown in Fig. 26. The vapor of the refrigerant enters at *A*. Cooling water enters

through the valve V and flows out through spray holes or narrow slots in the distributing pipe P, falling as a rain or spray over the surfaces of the rack of pipes below it. The vapor of the refrigerant when it condenses is drained off at the "bleeder" connections at B, C, and D. These bleeders are provided to remove

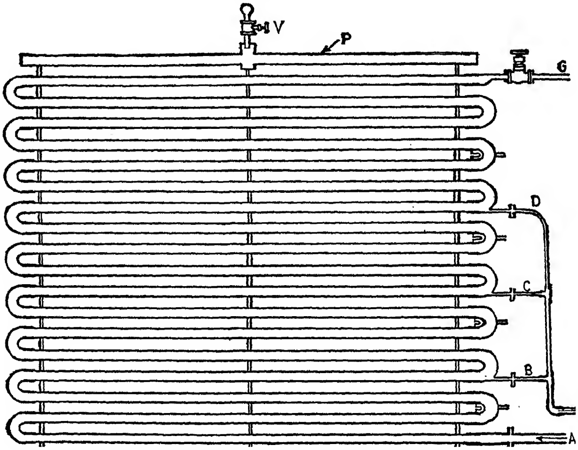


FIG. 26.—Atmospheric condenser (bleeder type).

the liquid refrigerant from the lower tubes of the condenser and convey the liquid to the liquid receiver. By constantly removing the liquid which is formed in the pipe coil, a greater amount of heat is transferred to the cooling water than if the liquid refriger-

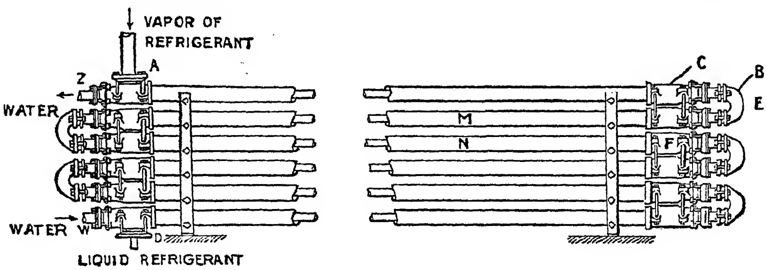
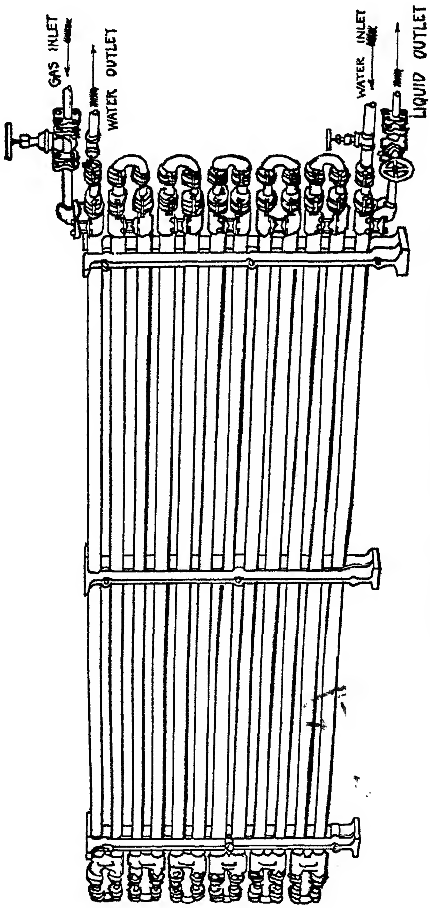


FIG. 27.—Double-pipe condenser.

ant is allowed to accumulate. If the liquid is not drained from the condenser, it will accumulate in the bottom pipes and reduce the effectiveness of the condensing surfaces.

Double-pipe Condenser.—The double-pipe type of condenser shown in Fig. 27 is ordinarily used in refrigerating plants where

the mist from falling water is objectionable and where there is little tendency for the cooling water to form scale. In this type of condenser, one pipe is placed inside another, and heavy fittings are used at the ends of the pipes so that cooling water may be



].—Double-pipe condenser showing piping connections.

passed through the inner pipe while the vapor of the refrigerant flows into and is condensed in the annular space between the two pipes. The vapor enters the top of the condenser through a special casting *A* which is provided with a stuffing box with a metallic packing not corroded by the kind of refrigerant used.

At the right-hand ends of a pipe coil, as at E, the water pipes are connected by a return bend B, while a casting C supporting the two sets of pipes at the bend is connected so that the vapor of the refrigerant passes through it to the next pipe level M and then, flowing toward the left, passes on into the next lower level N through a return bend and casting exactly like B and C. The water and vapor pipes are connected in this way until the drip box D is reached, from which the liquid refrigerant passes to the liquid receiver or other similar container. The cooling water enters at W and flows upward through the pipes, thus applying the counter-current principle. A slightly different design of double-pipe condenser is shown in Fig. 28.

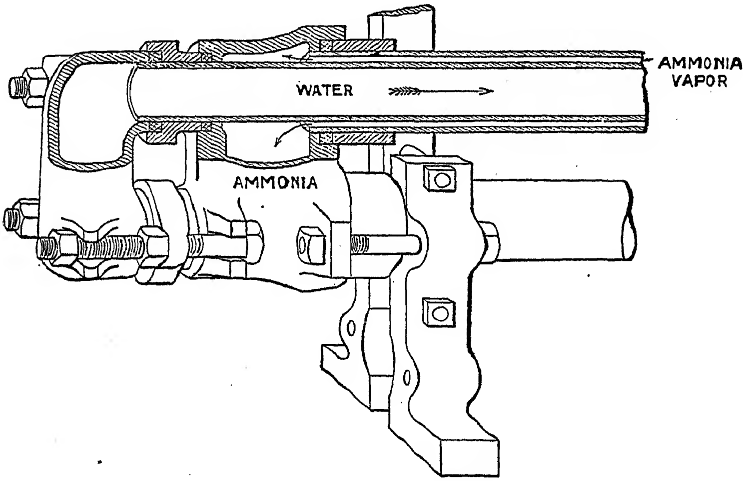


FIG. 29.—Details of double-pipe condenser.

Figure 29 shows clearly a detailed design of one make of return bend and casting for the ends of a double-pipe ammonia condenser. The directions of flow of ammonia vapor and cooling water are shown by arrows.

Attached to the top of a condenser is usually a blowoff or "purge" pipe, as shown at G in Fig. 26, which leads to a space provided for collecting the vapor of the refrigerant and is used to relieve the condenser of air and foreign gases which may accumulate. Air sometimes leaks into the system on the low-pressure side and is pumped around through the system into the condenser. This is apt to occur if low suction pressures are used. If air is

allowed to remain in the condenser, its accumulation will prevent the vapor of the refrigerant from entering all of the tubes of the cooling coil of the evaporator, and will thus reduce the capacity. As shown in Fig. 13, there is a connection to the blowoff pipe which extends upward from the liquid receiver. This piping provides for the removal of air or foreign gases which may accumulate in the liquid receiver. This connection is called the *equalizer line* and serves, also, to equalize the pressures in the liquid receiver and in the condenser, thus preventing any interruption in the flow of the liquid refrigerant from the condenser to the receiver.

The cooling surface of a double-pipe condenser is made only about one-third as large as the cooling surface of an atmospheric condenser. A double-pipe condenser with 1¼-inch inner pipes and 2-inch outer pipes, 19 feet long and 12 pipes high has a cooling surface of about 95 square feet and is suitable for an ammonia plant having a capacity of about 15 tons of refrigeration, allowing about 6.3 square feet of cooling surface per ton of refrigeration. The rate of flow of the cooling water should be about 275 feet per minute for the best results.

The most important advantages of the double-pipe condenser over some other types are that it makes full use of the counter-current principle, thus delivering the liquid refrigerant at the lowest possible temperature, and that it can be located nearly anywhere in the plant. It requires no condenser pan, with its possibility of wetting the surroundings.

The quantity of water required for a double-pipe condenser is the same as that required for an atmospheric condenser. This is because the double-pipe condenser does not have the advantage of the cooling effect produced by evaporation. This disadvantage is fully offset, however, by the operation of this condenser on the counter-flow principle. The double-pipe condenser may be located, in most plants, in the compressor room, where a trusty engineer can look after it, but it should not be placed in a warm room, as its capacity there will be reduced.

The quantity of cooling water at various initial temperatures which is required per minute for each ton of refrigeration in an ammonia plant, using *atmospheric* or *double-pipe* condensers is as shown in the table on page 46.

The values given in the table are based on water leaving the condenser at 95° F. The smallest sizes of pipe and amounts of water that should be allowed are given in the table on p. 51.

Initial Temperature of Cooling Water, Degrees Fahrenheit	Gallons per Minute
50	1/2
55	5/8
60	3/4
65	7/8
70	1
75	1 1/4
80	1 1/2
85	2

Shell-and-tube Condensers.—A kind of ammonia condenser which is coming into quite general use is the *shell-and-tube type*,

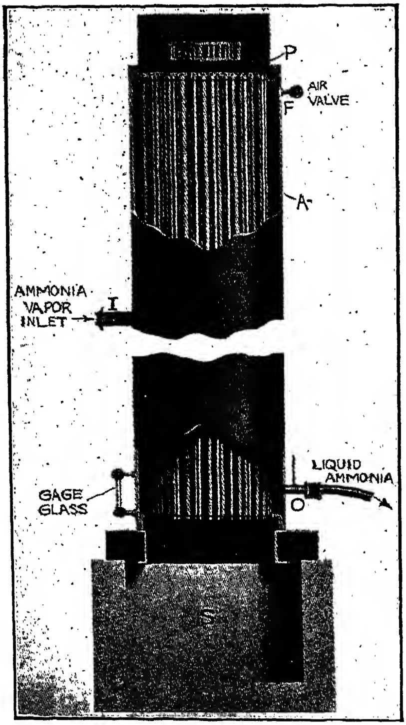


FIG. 30.—Shell-and-tube condenser.

which may be made in a number of different ways embodying, however, the same principles. The design most frequently found is shown in Fig. 30. It consists of a vertical cylindrical shell *A*

to the top of which a heavy tube sheet is welded or riveted. A number of steel tubes are set and preferably welded into holes in this tube sheet. The ammonia vapor enters the shell of the condenser at *I*. Cooling water enters the condenser through the water inlet pipe shown at the top of Fig. 31, and discharges into a so-called *water box* where uniform distribution of water is obtained by the use of a perforated plate and slotted ring distributors. Water from the inlet pipe at the top enters the inner slotted ring and is distributed by means of slots in the first and second of the concentric rings, so that it spreads out evenly over

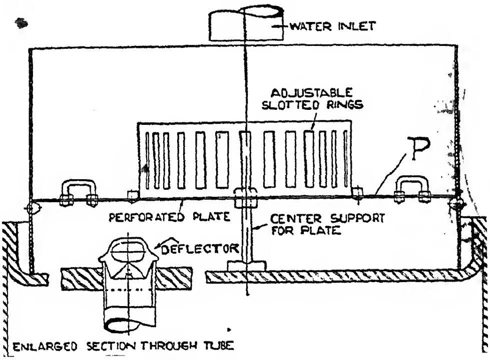


Fig. 31.—Details of shell-and-tube condenser.

the entire surface of the perforated plate *P* without damming up or surging. Two slotted distributing rings are used, one of which fits closely inside the other, and are arranged to make it possible to adjust the size of the effective opening through the slots. The slots in the two rings coincide when set for the maximum size of the opening for the discharge of water, and the rings are adjustable in position so that they may be moved to secure a "shutter" effect, thus obtaining a fine regulation of the openings to correspond to the amount of water to be discharged through them. The cooling water which passes through the perforations in the plate *P* is evenly distributed around the tops of the water tubes which are in the body of the shell. The tops of these water tubes are provided with cast-iron deflectors intended to distribute a film of water over the surface of the inner walls of the tubes. The cast-iron deflector on each of the tubes has a handle so that it may be easily removed when the tube is to be cleaned. There are also handles on the perforated plate so that this plate together

with the slotted rings may be easily removed for cleaning. The water, after passing through the tubes, falls into the sump or pit S (Fig. 30), from which it discharges into a drainpipe or sewer.

In the operation of this condenser, the ammonia vapor enters through the inlet pipe I and is condensed on the surfaces of the tubes. The liquid ammonia which is formed by condensation collects in the bottom part of the shell and is drained off through the liquid outlet pipe O. There is an air-purge valve F near the top of the shell.

The cast-iron deflectors which are set into the tops of the tubes are sometimes made with a cylindrical shape instead of conical as shown in Fig. 31, and have spiral grooves on the cylindrical surface. The object of providing these spiral grooves is to give the flow of water a corkscrew motion down through the tubes. The condenser tubes may be cleaned by simply removing the

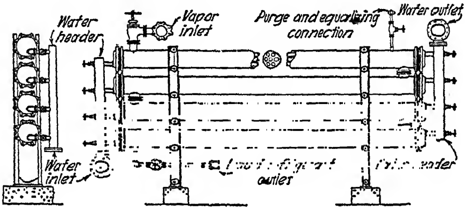


FIG. 32.—Multipass condenser.

perforated plate P and the cast-iron deflectors from the tops of the tubes and then working a tube cleaner up and down in the tubes. When installing a condenser of this type, headroom should be provided, so that the tube cleaner may be used without difficulty.

The shell-and-tube type of condenser has a practically parallel flow of water and ammonia vapor and does not, therefore, have the advantages of the counter-current principle. This kind of condenser, however, will usually cool the liquid ammonia to within 4 to 8° F. of the temperature of the cooling water, when it is discharged from the bottom of the condenser. It requires a comparatively small amount of tube surface per ton of refrigeration, due to the especially good heat transmission through the tubes in this design.

Multipass Condenser.—A type of condenser which is intended to combine the advantages of both the double-pipe type and the shell-and-tube type is shown in Fig. 32, and is called a "multi-

pass" condenser. It is constructed in its essential parts of a number of very large pipes usually 8 to 20 inches in diameter, each of these large pipes containing a number of 2-inch tubes. A commonly used design with seven 2-inch tubes in each pipe is shown in the figure. At each end of one of these units there is a "water head" properly baffled as shown in Fig. 33, so arranged that the water will pass from end to end of each one of the seven 2-inch tubes before it discharges from the outlet of the unit. These units are generally arranged in vertical tiers with one unit above the other as shown, to form a "stand." The inlet for the vapor of the refrigerant is at the top of the condenser. It passes from right to left from one unit to the next until it discharges through the liquid refrigerant outlet at the bottom of the lowest unit.

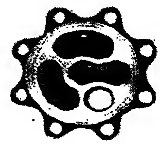


Fig. 33.—Water-head of multipass condenser.

The advantages claimed for this type of condenser are: (1) infrequent repairs; (2) simple construction; (3) small space; (4) low resistance to liquid flow, so that the cooling water can be passed through other units after discharging from one of this kind without being pumped again; (5) small frictional resistance to the flow of vapor.

Flooded Condensers.—Both atmospheric and double-pipe condensers may be made so that there is some liquid ammonia on the bottom surface of one or more of its pipes. A condenser designed for this kind of operation is called a *flooded condenser.* It is the theory of this design that if the vapor of the refrigerant as it comes from the compressor into the condenser is mixed with a small amount of liquid ammonia, the mixture of vapor and liquid ammonia will be condensed by the cooling water supplied to the condenser at a higher rate than if the pipes were filled with only ammonia vapor. There is a lower rate of heat transfer from ammonia vapor to water, especially when the ammonia vapor is superheated, than from a *mixture* of ammonia vapor and liquid ammonia to water. The injection of liquid ammonia directly into the pipes of an ammonia condenser by means of an ejector nozzle, as shown at B in Fig. 34 and in detail at the right-hand side of the figure, has obviously the effect of removing, or at any rate reducing, the amount of superheat in the ammonia vapor as it comes from the compressor. Recent tests, however, have demonstrated conclusively that there is no merit in this

somewhat expensive type of condenser construction, and that the best kind of condenser is the one which drains the condensed ammonia most promptly from *all* the piping in the condenser.

In the flooded type of atmospheric condenser, as shown in the figure, the lower pipes operate in practically the same way as those in an ordinary atmospheric condenser, and it is likely that after the liquid ammonia which discharges through the ejector is picked up by the ammonia vapor, it is carried more or less like slugs of liquid and vapor to the upper pipes where the liquid separates from the vapor.

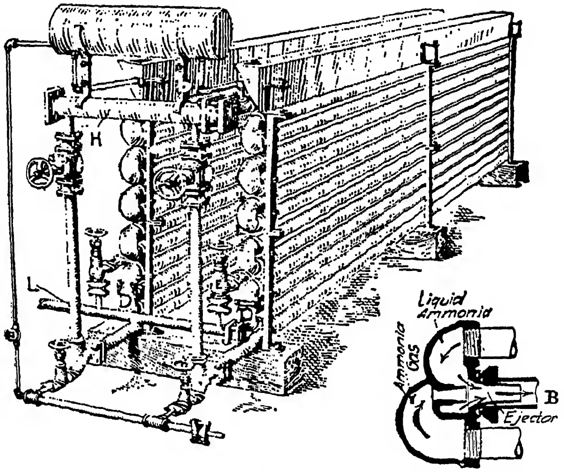


Fig. 34.—Ammonia condensers arranged for operation on "flooded" principle.

It is difficult to operate several flooded condensers in parallel from a common header *H*, as shown in Fig. 34, as it is almost impossible to keep all the condensers operating under the same conditions. There is the further disadvantage that a flooded condenser cannot be operated with as low pressures in the condenser as the ordinary types. It is not unusual for the pressure in a flooded type of condenser to increase a considerable amount without any apparent reason. This type was formerly used to a considerable extent, but is now practically obsolete.

Selecting Ammonia Condensers.—Until recently, the atmospheric type of condenser was very much in favor, but a preference for the shell-and-tube and multipass types are observed in some of the recent installations. This is especially true where cooling

towers are not to be used and water must be pumped from a low level. In that case, the shell type of condenser can be located in a place where the elevation to which water must be pumped is small, while, on the other hand, atmospheric condensers must usually be placed on the tops of buildings, where the additional pressure due to elevation is considerable.

The following table gives the surface and cooling water required by different types of *ammonia* condensers. It is based on a condenser-gage pressure of 200 pounds per square inch and on a cooling-water temperature of 70° F.[1]

TABLE I

Type of condenser	Cooling surface per ton of refrigeration, square feet	Transmission, B.t.u. per square foot per minute per ton	Pipe, per ton of refrigeration. Size most used, inches	Lineal feet of pipe per ton of refrigeration	Water, gallons per minute per ton
Flooded double pipe	5	40.0	1¼	14	1½–2
Flooded atmospheric	6	33.3	2	11	1½–2
Double pipe	8	25.0	1¼	22	1½–2
Shell and tube	18	11.0	1½	55	2 –3
Atmospheric	24	8.3	2	45	1½–3
Submerged	35	5.7	2	65	3 –7

Condensers cannot be efficient unless clean. Oil, scale, and dirt gradually accumulate and tend not only to clog the pipes but also lower the heat conductivity.

Close attention to the refrigerant and the cooling-water temperatures will give an accurate indication of the condition of the condenser. Increased condenser pressure is another indication, although conditions will have become very bad before it is noticeable. High condenser pressure may indicate the accumulation of non-condensable gases, dirty tubes, too high condensing-water temperature for a given cooling surface, an excessive amount of ammonia in the system, or insufficient cooling water.

[1] Complete data of tests to determine the heat transfer in various types of ammonia condensers are given in *Ill. Eng. Exp. Sta. Bull.* 171, 186, and 209, which give the results of tests by Kratz, MacIntire, and Gould.

A high condenser pressure means an increase in power, greater wear and tear on apparatus, increased liability to leaks, and forced shutdowns. Because of this, the condenser pressure should be closely watched and should be kept as low as possible.

Water-regulating Valve.—The automatic control of the water supplied to the condenser is usually taken care of by a valve which opens at a definite condenser pressure. This pressure is exerted on a diaphragm. The principle of operation of this valve is shown in Fig. 35. The space above the diaphragm is connected to the condenser. Any increase in condenser pressure deflects the diaphragm which opens the valve. When the compressor stops, the water flowing over the condenser lowers the condenser pressure below the normal value and closes the water valve. A strainer should be placed "ahead" of the valve to prevent its being closed by the accumulation of sediment and other foreign matter when the compressor is not operating.

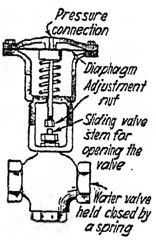


Fig. 35.—Automatic water valve.

Water-cooling Systems.—It often happens that a plant is so situated that condenser cooling water must be purchased from the city. As ice plants require from 100 to 800 gallons per minute of cooling water, it is readily seen that the greatest economy should be attained. In order to save the water and to use it again for cooling purposes, several methods are in use to cool it; namely, cooling ponds, sprays, and cooling towers.

Cooling Ponds.—In this method, the water is partly cooled by radiation and conduction but chiefly by evaporation. The normal condition of the air is such that it can readily absorb more water vapor. Its capacity for absorbing water vapor is increased by contact with warm water and by radiation. Cooling ponds generally require considerable space, and their use is often impracticable. In such a case, cooling sprays are used to accomplish the same object.

Cooling Sprays.—With this method, the hot circulating or cooling water is distributed through pipes and discharged from nozzles into the air, falling like a heavy rain into a pond. The nozzles are so designed as to cause the discharged water to separate into drops. The water issuing from the nozzles induces a draft which, aided by the natural breeze, increases the evaporation. Sprays are often located on the roof of the plant. The

loss of water carried off by the air seldom exceeds 4 per cent. In general, under ordinary conditions, the power required to operate the spray will average less than 1.5 per cent of the power required to operate the plant.

Cooling Towers.—A cooling tower is made up of a wooden or sheet-iron housing. This housing is open at the top and at the bottom and is arranged so that the hot water may be raised to the top of the tower. The water is then distributed in such a way as to cause it to fall in thin sheets or sprays into a reservoir at the bottom. Air at the same time is drawn in at the bottom by natural draft or forced in by a fan. The water in falling gives up its heat, chiefly to the rising air, by evaporation and convection.

Water-cooling towers are built in several different types as follows: (1) those with forced draft; (2) those with natural draft, open to the atmosphere; (3) those with natural draft closed to the atmosphere; and (4) those with combined natural and forced draft. Forced-draft towers are completely enclosed, except at the top and at the bottom, where space is left for the fan to open. In the natural-draft tower, the sides are lowered, and the necessary air is supplied through the open base and through the lowered sides by natural-air currents. The natural-draft closed type is like a chimney in its action. The combined forced and natural-draft tower may be used with natural draft for light loads and with forced draft for heavy loads.

Liquid Receiver.—A gage glass (Fig. 180) similar to those on oil separators is generally attached to the liquid receiver to indicate the amount of liquid, as it is important that the supply of liquid refrigerant should not be too large or too small. If a large quantity of liquid refrigerant should collect in the liquid receiver, the liquid will flow back into the condenser; on the other hand, if the quantity becomes small, there may not be enough liquid to supply the cooling coils of the evaporator. The liquid receiver in an ammonia refrigerating system serves, to some extent, as an oil separator, because the oil, being heavier than the liquid ammonia, sinks to the bottom of the receiver from which it can be drained.

In case the refrigerating plant is to be shut down and all of the liquid refrigerant in the system is to be stored in the liquid receiver, it should be about twice as large as the ones ordinarily used. Valves should be placed in the inlet and outlet connec-

tions of the liquid receiver. The liquid receiver of an ammonia refrigerating system is usually large enough to hold about ½ gallon of liquid ammonia for each ton of refrigeration capacity.

Scale Separator.—After the vapor of the refrigerant leaves the cooling coils of the evaporator, it should pass through a scale separator to remove scale which may have been freed from the inner walls of the coils and to prevent scale from finding its way to the compressor, where it would damage the valves.

Ammonia-vapor Precooler.—An ammonia vapor precooler consists usually of a metal shell containing a coil in which the liquid ammonia circulates. It is the practice of one designer to have the cold water enter at one end and pass out at the other. The ammonia vapor from the evaporator enters on one side and, after circulating through the space between the coil and the shell, passes out on the other side. At the bottom, there is an oil leg, so that this device serves as an oil separator as well as an ammonia-vapor precooler. Recent tests made of such a precooler show that a plant of 50-tons ice-making capacity operating at 25 pounds per square inch suction-gage pressure and 150 pounds per square inch discharge-gage pressure will heat 4.62 gallons of water per minute when the temperature of the discharged ammonia vapor is 217° F., the temperature of the water entering the precooler being 65° F., and the temperature of the water leaving the precooler 95.3° F. For this temperature range (30.3° F.), the heat transfer coefficient is 26.3 B.t.u. per hour per degree mean difference in temperature per square foot of surface. Ammonia-vapor precoolers are not extensively used, because it seems unwise to employ two pieces of apparatus to perform the same purpose. In other words, the condenser is depended on to remove heat from the ammonia vapor discharged from the compressor. The most useful application of precoolers, however, is in refrigerating plants in which ice is being made, the water for ice making being passed through a precooler to be heated for use in the hot-water tanks in which the ice cans are dipped, to facilitate the removal of the ice from the cans.

The more common method of precooling the liquid ammonia is by the use of the *accumulator* to be described in connection with Figs. 47 and 48, in which the liquid ammonia is cooled by the ammonia vapor in the suction line of the compressor. By this means the liquid ammonia is cooled to nearly the temperature in the evaporator.

Enclosed Compression Unit.—An unusual type of refrigerating machine in which the compressor, condenser, brine cooler, and pipe system are in a single unit is shown in Fig. 36. The Audiffren-Singren refrigerating system, made for the H. W. Johns-Manville Company, is contained in two nearly spherical chambers A and V. There is a hollow shaft S, supporting a bowl-shaped casting B which is kept from turning by a heavy weight W. This casting carries the trunnions T, T of a cylinder C, the piston P of which is connected to a rod attached to the strap of an eccentric D on the shaft S. The shaft S is driven by means of a belt on the pulley Q. The circular chamber A and the oval

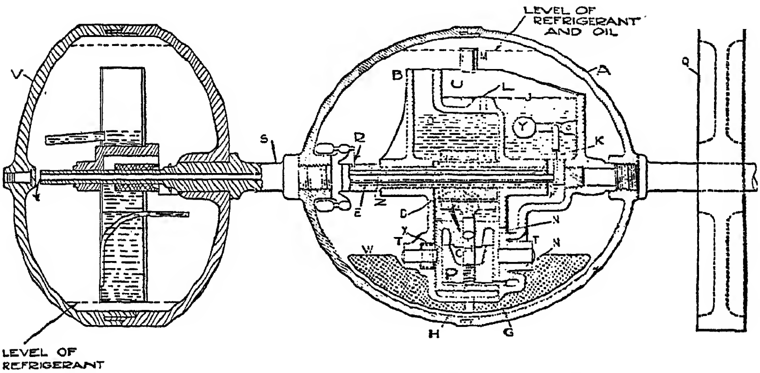


FIG. 36.—Audiffren-Singren refrigerating apparatus.

chamber V are revolved by the shaft S. When the shaft rotates the chamber A, the cylinder C, which is a part of the large chamber B, remains nearly stationary, *oscillating* only a little under the influence of the heavy weight W, which causes it to hang down. The piston P in the cylinder C is drawn in and out of the cylinder by the eccentric D. The cylinder C oscillates between the "faces" of the suspended bowl-shaped casting B on the trunnions T, T. The right-hand face of the cylinder C is pressed by the spring X against the face of the lower part of the casting B which contains ports or holes marked N.

In this way, ports or holes in the two ends of the cylinder C are connected to the two suction ports N, N in the lower part of the casting B at the proper times in somewhat the same way that the distribution of steam is brought about in the cylinder of a reciprocating engine. Thus, the vapor of the refrigerant (sulphur

dioxide) is admitted to the cylinder *C* at *F*, from the annular space *E* between the two hollow shafts. When the port or hole at *G* in the cylinder *C* and in the "face" at *N* come opposite, the vapor of the refrigerant is admitted into the cylinder and is compressed. When the proper pressure is reached, the cylinder discharge valves at *H* open and discharge the vapor into the casing *A*. Now, the casing *A* *revolves* (partly immersed) in a tank containing water for cooling, which condenses the refrigerant and this liquid together with the oil supplied for lubrication collects by centrifugal force at the outer part of the revolving casing *A*, where some of it is caught up by a scoop *M* and is collected in the reservoir *J*. After the lubricating oil is removed, some of the liquid refrigerant passes through an expansion valve *K* which is regulated by a suitable float *Y*.

The oil floats in the reservoir *J* and overflows through the hole *L* into the chamber *O*, in the lower part of which the cylinder *C* oscillates, so that the eccentric *D* and the piston *P* are flooded with oil. The whole interior is under pressure, so that there is no leakage of refrigerant from the chambers. There is a tendency for the oil to enter around the piston rod and between the valve faces; but the spring *X* holds the lower part of the casting *B* against the sliding face containing the port or hole *N*. The liquid refrigerant which is at low pressure after passing the throttling expansion valve *K* passes along the inside of the inner shaft extending between the two chambers *A* and *V* and finally settles, by centrifugal force, at the circumference of the oval-shaped chamber *V*. In the low-pressure chamber *V* (evaporator), the liquid refrigerant is vaporized as it removes heat from brine in a tank in which it revolves.

The vapor of the refrigerant returns from the chamber *V* to the chamber *A* through a space (not shown) formed between the two hollow shafts which connect the chambers *V* and *A* and then passes again into the compressor through the annular space *E* and the opening *F*. The complete system is contained within a tight set of nearly spherical chambers and shafts, so that there are no moving joints to be kept tight, and, consequently, there is no danger of leakage. An extension on the right-hand chamber serves as one journal for the system, and the hollow shaft at *S* serves as the other. There is little weight on the journals, as the buoyancy due to the two chambers *A* and *V* being immersed supports much of the weight. The pressure of the vapor of the

refrigerant in the chamber A, in addition to the spring pressure, tends to keep tight the sliding joint between the oscillating piston P and the "face" of its cylinder. Should the cooling water be shut off and the temperature rise, the high pressure developed would finally be sufficient to cause the weight W to rotate and so prevent a further rise in pressure. The small valve at R is held down, when the apparatus is in operation, by centrifugal force of the ball weights, as shown, but upon stopping the machine, this valve is opened by the weight of the balls, thus equalizing the pressure.

The cooling water in a tank below the chamber A *condenses* the vapor of the refrigerant while the *evaporation* of the liquid refrigerant in the chamber V cools the brine in another tank. When the brine in this tank is cooled to approximately the temperature of the vapor of the refrigerant, there will be no further evaporation of the liquid refrigerant, no vapor will be sent back to the compressor in the chamber A, and, consequently, no vapor will be liquefied in the chamber A. After a short time of operation of the apparatus, the level of the liquid in J will close the float valve K, and no more liquid refrigerant can pass over to the evaporator in the chamber V.

Compound Ammonia Compression System.—It has usually been considered good engineering practice to install an ammonia absorption refrigerating system when a low temperature was to be maintained in the cold-storage rooms or when the suction- or inlet-gage pressure was less than 10 pounds per square inch. There are, of course, many single-cylinder ammonia *compression* systems which are operating with a lower suction pressure than this value. It is generally admitted, however, that the single-cylinder ammonia compressor does not give satisfactorily economical operation with such low suction pressure, and the ultimate plan of operation for best economy is either to install an absorption refrigerating system or use a compression system which is equipped with a compound (two-stage) compressor. The compound ammonia compression system is, of course, more complicated than the system requiring only a single-cylinder compressor. The compound compression system requires, in addition to the usual equipment of the ordinary compression system, the following apparatus: (1) high-pressure compressor cylinder; (2) low-pressure compressor cylinder; (3) low-pressure discharge vapor cooler; and (4) intermediate receiver.

In the operation of the compound compression system, the low-pressure cylinder of the compressor takes in the ammonia vapor from the coil of the evaporator through its suction pipe. This vapor is compressed to an intermediate pressure in the low-pressure cylinder, from which it is discharged into the ammonia-vapor cooler. The vapor cooler is provided for the purpose of removing any superheat in the ammonia vapor. The cooling in this apparatus is usually done by circulating water. From the vapor cooler, the ammonia vapor passes on to the intermediate-pressure receiver. Part of the liquid ammonia which is

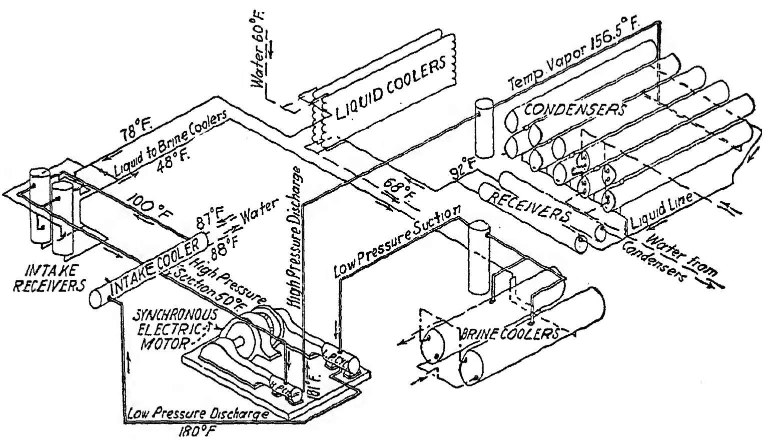


FIG. 37.—Compound ammonia compression plant.

obtained by the condensation of ammonia vapor in the condenser is also discharged into this intermediate-pressure receiver, and when thus discharging it immediately expands and vaporizes. The suction pipe of the high-pressure cylinder of the compressor opens at one end into this intermediate-pressure receiver, so that the ammonia vapor in the receiver is drawn into the high-pressure cylinder. The discharge of the high-pressure cylinder is into the condenser.

The liquid ammonia which accumulates in the bottom of the intermediate-pressure receiver or trap is transferred through a pipe into the coils of the evaporator where it expands. The expansion or regulating valve is provided in this pipe to control the pressure in the coil of the evaporator. Two principal advantages are claimed for the compound compression system, and these are: (1) less horsepower required for compression and (2)

a lower discharge temperature in the condenser than with a system having a single-cylinder compressor.

Figure 37 shows the layout of a compound ammonia compression plant. The compressor as shown is driven by being directly connected to a synchronous electric motor which operates at 150 revolutions per minute. The stroke of the cylinder of the compressor is 24 inches, and the diameter of the high-pressure cylinder is 13 inches, that of the low-pressure cylinder 22 inches.

A compound compressor, especially if of the *rotary* type, is sometimes called a *two-stage* compressor. There are also *multistage* compressors having a larger number of stages, as, for example, the compressor described on page 132.

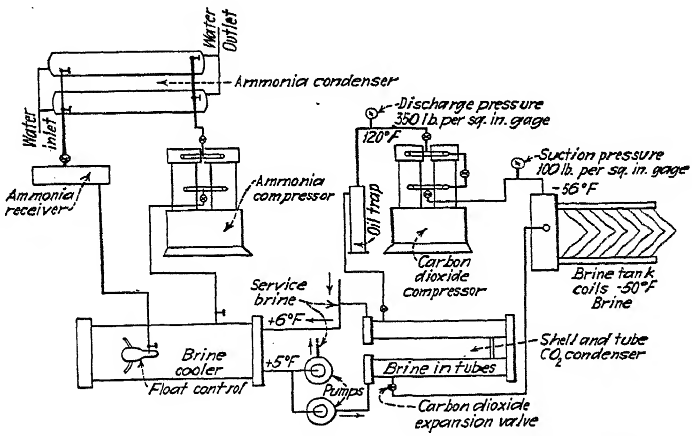


FIG. 38.—Combined ammonia and carbon-dioxide refrigerating system.

Combined Ammonia and Carbon-dioxide System. Binary System.—To obtain the extremely low temperatures required in recently developed systems of refrigeration (especially for "quick freezing," p. 431), the advantages of the carbon-dioxide system have been combined with the ordinary ammonia system to obtain the required results. Such a system utilizing two different refrigerants is called a *binary system.* In its application there are required an ammonia compressor and its condenser, as well as also a carbon-dioxide compressor and its condenser. In the operation of the system, the discharge pressure of the carbon-dioxide unit is lower than the discharge pressure which would ordinarily be used when water is the cooling medium in the condenser. By the use of this abnormally low discharge

pressure, the power required to operate the system is much less than it would be for a higher discharge pressure, with the result that the operation of the system is made more economical by this means.

Figure 38 shows diagrammatically, somewhat in detail, this combined system with the ammonia compressor and its condenser on the left-hand side and the carbon-dioxide compressor and its condenser on the right-hand side. The ammonia unit utilizes water for cooling its condenser, and the ammonia refrigerant cools brine in a brine cooler, and this cooled brine is used as the cooling medium in the carbon-dioxide condenser. There is also a brine tank which is cooled by the carbon-dioxide refrigerant.

The *heat removed* in this "carbon dioxide" brine tank is *carried over* to the carbon-dioxide condenser where it is transferred to the brine from the ammonia condenser, and in this latter condenser the heat is absorbed by the ammonia refrigerant which in turn gives up this heat to the cooling water passing through the ammonia condenser.

Refrigerating System for Air Cooling.—When refrigeration is required for cooling the air in connection with air conditioning for halls and rooms in buildings, the refrigerating or cooling surfaces of the evaporator and the compressor are usually made very compact, to take up as little floor space as possible. Since nearly all air-cooling installations require the use of air at relatively high temperatures, it is possible, of course, to operate the compressor of the refrigerating system with a comparatively small difference between the discharge and the suction pressures. Because of this fact, the refrigerating capacity of a compressor in this kind of service is usually about 100 per cent more than the standard rating.

A self-contained type of air-cooling system manufactured by the Carrier Engineering Corporation is shown in section in Fig. 39 and pictorially in Fig. 40. The nozzles *N* (see Fig. 39) for spraying the re-circulated liquid refrigerant over the brine tubes *A*, which are to be cooled, are located in the left-hand chamber, while the tubes *C* for circulating the cooling water in the condenser are on the right-hand side. The compressor, which is of the centrifugal type (p. 132), is located above and between the brine tubes *A* and the condenser tubes *C*. The liquid refrigerant, after passing through the spray nozzles *N*, is vaporized and drawn into the suction or inlet of the centrifugal compressor *M*

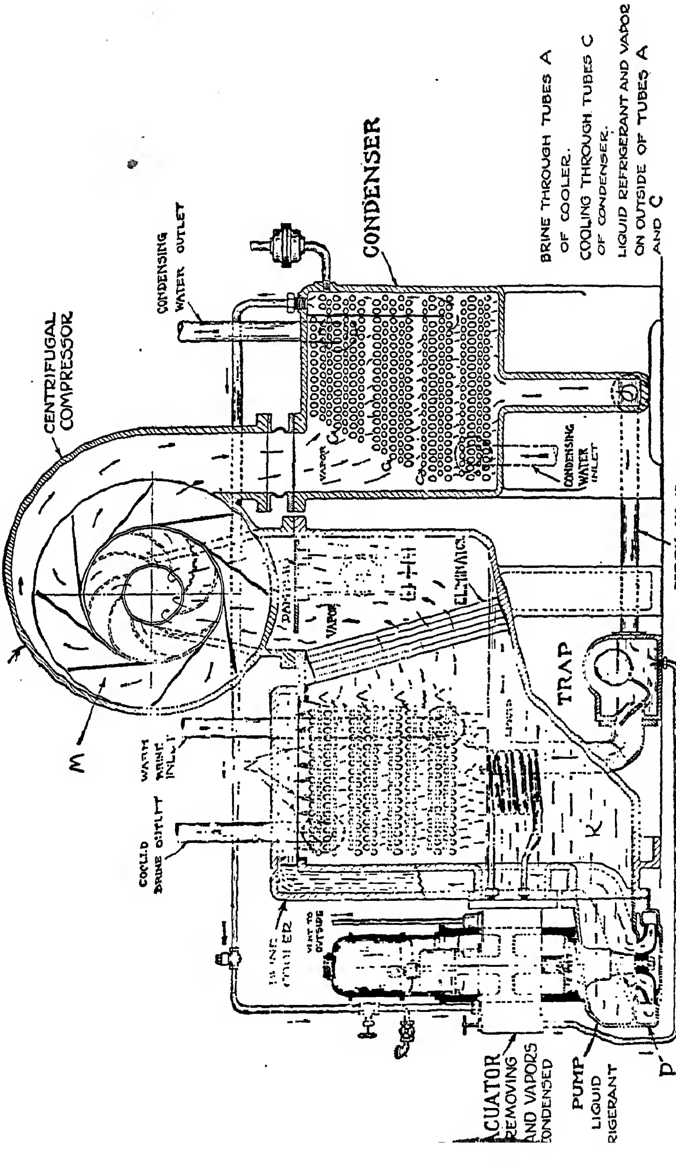


FIG. 39.—Diagrammatic arrangement of centrifugal compression system for air cooling.

where, after being compressed to a higher pressure, it passes into the condenser tubes C. In the condenser, the vapor of the refrigerant becomes a liquid and flows by gravity into the liquid reservoir R at the bottom of the apparatus from which it is re-circulated through the system by the motor-driven centrifugal pump P located at the side of the apparatus.

Figure 40 shows a plant of this kind which is operated by an electric motor and reduction gearing. The centrifugal com-

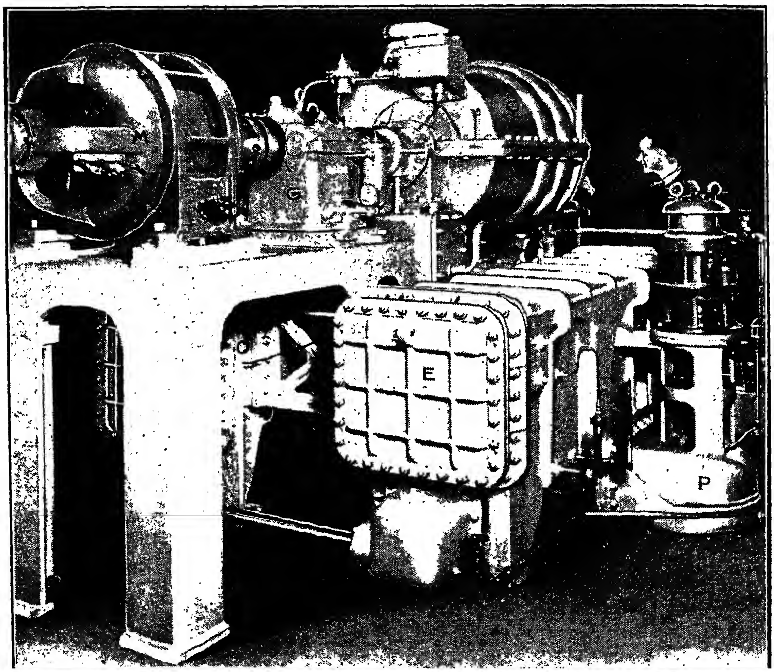


FIG. 40.—Motor-driven centrifugal compression system.

pressor used in the Carrier system for air cooling is shown in Fig. 85 (p. 132). This compressor is made with five compartments which are called "stages," and operated in most cases at about 3,600 revolutions per minute. At this speed it is well adapted to be operated by direct drive by a steam turbine. When such a high-speed compressor is driven by an electric motor, the driving power must be transmitted from the motor to the compressor by means of suitable gears because of the low speed of an electric

motor. The motor *M* drives the centrifugal five-stage compressor *C* through the gear box *G*. The vapor of the refrigerant is discharged at high pressure by the compressor into the condenser *O* from which the liquid refrigerant flows by gravity into the reservoir adjoining the pump *P* by which it is discharged through nozzles over the brine-cooling coils of the evaporator or "cooler" *E*, the pressure being reduced at the same time, so that nearly all the liquid is vaporized and cools (absorbs heat from) the brine. After evaporation, the vapor is drawn into the suction line leading to the compressor to be re-circulated in the system.

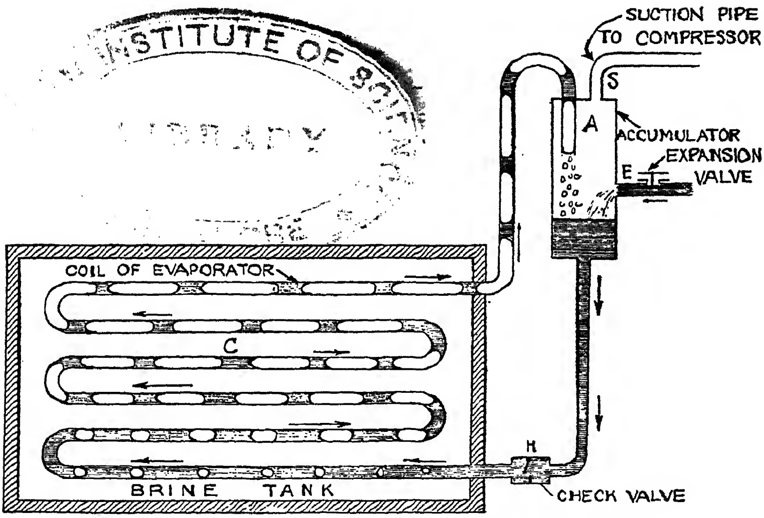


FIG. 41.—Diagram of "flooded" operation of evaporator.

Data regarding several refrigerants which are especially suitable for use with centrifugal compressors, where small differences between discharge and suction pressures are permissible, are given in the table on page 93.

Flooded System of Operation of Evaporator.—In Fig. 41, the cooling coil *C* of the evaporator of a refrigerating system is shown immersed in a brine tank. Between the expansion valve *E* and the coil of the evaporator is an accumulator *A* into which the liquid refrigerant discharges after passing through the expansion valve. From the accumulator, the liquid refrigerant flows by

gravity[1] into the bottom of the coil of the evaporator. The low-pressure end (top) of the coil *C* of the evaporator is connected up by piping so that the vapor of the refrigerant discharges into the top of the accumulator *A* where it loses any entrained drops of liquid refrigerant, and only perfectly dry vapor passes out into the suction pipe *S* to the compressor. The pressure in the accumulator is nearly the same as the suction pressure at the compressor. The warm liquid refrigerant entering the accumulator from the expansion valve is cooled in the accumulator to the saturation temperature corresponding to the suction pressure. A *check valve H* is in the liquid line to prevent any vapor of the refrigerant which may be formed in the lower part of the coil of the evaporator from passing back through the liquid into the accumulator. In the application of this apparatus, the refrigerant does not become entirely vaporized in the coil *C* of the evaporator but is distinctly a *mixture* of the liquid and the vapor of the refrigerant, which is discharged into the top of the accumulator.

The use of the descriptive term *flooded* in connection with this method of operating the evaporator is to express, in a way, the condition of the refrigerant in the coils of the evaporator, which, however, are not, strictly speaking entirely "flooded." The mixture of the liquid and the vapor of the refrigerant in the coil of the evaporator may be represented by relative amounts of liquid and vapor, as shown roughly in Fig. 41. The expansion valve *E* must be regulated so that the liquid level is maintained at a fairly constant elevation in the lower part of the accumulator *A*.

The principal advantage of the flooded system of operation of the evaporator is its simplicity, especially in a plant where there are a number of evaporator coils which may be connected to single liquid and suction headers connected into a single accumulator, thus reducing considerably the number of valves requiring adjustment. This system, further, regulates automatically the flow of refrigerant to each coil of a large evaporator, supplying the refrigerant in proportion to the amount required.

The system is effective in eliminating the drops of entrained liquid refrigerant which may be mixed with the vapor when it leaves the coil of the evaporator, so that, as a rule, the vapor of

[1] Because the flow of liquid refrigerant from the accumulator into the coil of the evaporator is due to gravity, this system is sometimes called a *gravity-feed system*.

the refrigerant enters the suction valves of the compressor very nearly saturated or only slightly *superheated.* This condition of the vapor permits the compressors in a compression refrigerating plant to operate under the most favorable conditions. Also, because the interior surfaces of the evaporator are partly covered with liquid, there is unusually good heat transmission, for the reason that heat transmission from liquid to liquid is much better

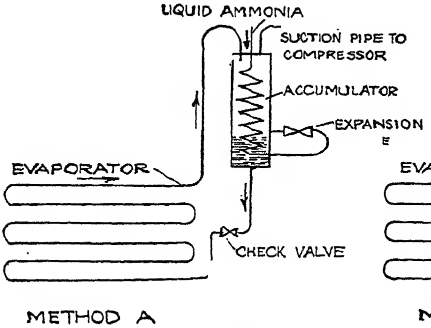
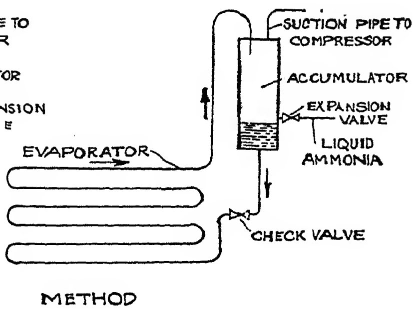
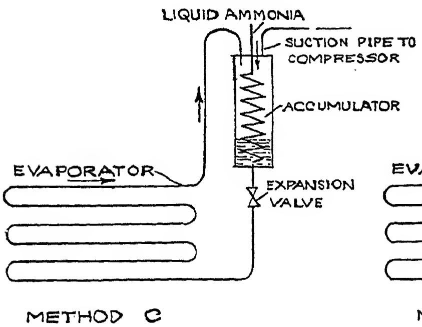


Fig. 42.



G. 43.



G. 44.

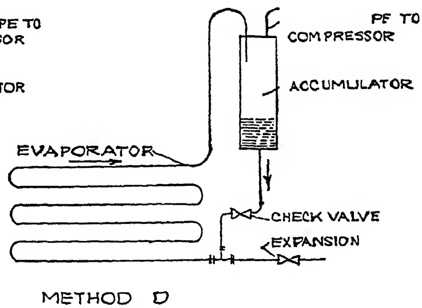


Fig.

Figs. 42–45.—Piping arrangements for flooded systems of operation of evaporator.

than from vapor to liquid. Better heat transmission in an evaporator makes possible the use of a smaller amount of coil surface than would otherwise be needed; or, without a change of the amount of coil surface, the compressor can be operated at a higher suction pressure. The important *disadvantage* of this system of operation of the evaporator is that a larger amount of refrigerant is needed to charge the system than when the liquid refrig-

erant passes directly from the expansion valve into the coil of the evaporator.

Four different methods of connecting the accumulator to the coil of the evaporator are shown in Figs. 42, 43, 44, and 45. These different methods are marked with the letters *A*, *B*, *C*, and *D*. In method *A*, the warm liquid refrigerant enters the top of a coil in the accumulator and, after passing through the expansion valve at the right-hand side of the figure, discharges into the liquid in the bottom of the accumulator. By this method, the liquid refrigerant flows by gravity into the coil of the evaporator. If the expansion valve is not properly regulated, the vapor leav-

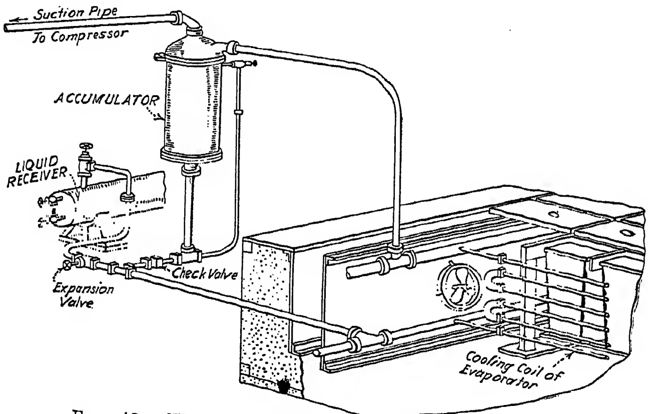


FIG. 46.—Flooded system using method *D* (Fig. 45).

ant passes directly from the valve i the coil of the
evaporator without first . The liquid
refrigerant which is vapor in the

accumulator falls into the accumulated liquid in the bottom of the accumulator and passes by gravity flow back into the coil of the evaporator. A check valve is placed in this "gravity-flow" line to prevent any possible flow of the low-pressure liquid from the expansion valve into the bottom of the accumulator.

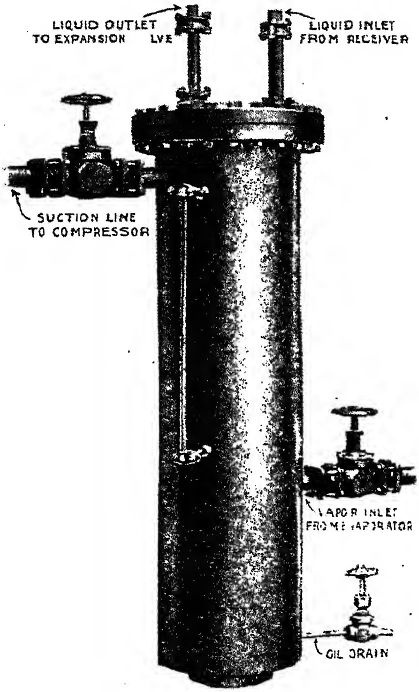


FIG. 47.—Accumulator for "flooded" system.

The flooded system of refrigeration is especially advantageous for use where low temperatures are constantly required. For quick or "*sharp*" *freezing,* a direct-expansion arrangement of *flooded* cooling coils (without the use of brine) has found many important applications. Such quick or sharp freezing is desirable for hardening ice cream, making ice, and other low-temperature work. Figure 46 shows pictorially the Frick flooded system, which is an application of method *D*. The York accumulator for a flooded system of operation of the coils of the evaporator

is shown in Figs. 47 and 48. This device operates by a variation of method C.

Float Valve for Flooded Systems.—A type of automatically operated expansion valve which is different in principle from the valves explained on page 15, is required for controlling the admission of the liquid refrigerant to the coils of the evaporator operated by the *flooded system*. This type being opened and

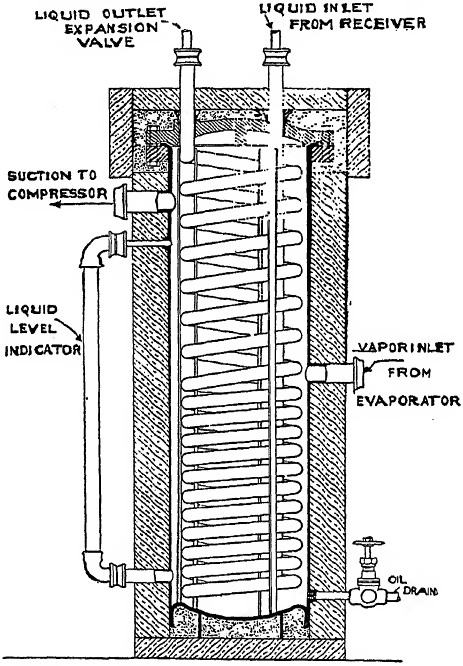
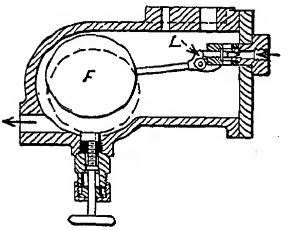


FIG. 48.—Cross-section of accumulator for "flooded" system using method C (Fig. 44).

closed by the action of a float is called a *float valve*. When the level of the liquid refrigerant in the coil of the evaporator reaches the level at which best operation is obtained, the float on the valve closes and stops the flow of the liquid refrigerant into the evaporator. As shown in Fig. 49, the float is marked F, and the lever L controls the needle valve operated by it. The elevation at which the float operates the valve can be adjusted by means of the screw shown at the bottom of the figure. The general arrangement for the use of a float valve in the *commercial* type of

refrigerating plant is shown in Fig. 50, the liquid refrigerant entering the coils of the evaporator at the bottom and its vapor discharging at the top through a *trap* which is a device arranged with a baffle plate *p* on which drops of the liquid refrigerant will collect because of the change in the direction of flow of the vapor around this plate. The condensation accumulating in the trap is discharged from the bottom through a pipe which enters the top of the float chamber. Both the float chamber and the trap are connected by suitable piping to a back-pressure *relief valve* as shown, which is spring-operated and intended to open if at



49.—Float valve for flooded system.

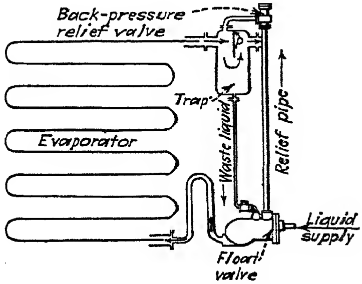


Fig. 50.—Commercial type of flooded cooling system.

any time an excessive pressure develops either in the float chamber or in the trap.

A float valve is applied to some types of small household and other refrigerating systems that are operated on the flooded system by the method of locating the valve so that the float chamber is on the high-pressure side of its needle-pointed expansion valve. The float chamber illustrated in Figs. 49 and 50 is, of course, on the low-pressure side of the expansion valve which it controls. When, therefore, the float chamber is on the high-pressure side of the valve it controls, its operation is necessarily just the opposite in effect of that of the valve already explained and is briefly as follows: When all the refrigerant has passed out of the *liquid receiver*, the float drops in its chamber to such a position that it closes the needle-pointed valve controlling the level of liquid refrigerant in this receiver. The valve controlled by the float then remains closed until the level of the liquid refrigerant accumulating in the receiver is high enough to raise the float sufficiently to open the valve. A float valve

used in this way is intended to make a "seal" between the high-pressure and the low-pressure sides of the refrigerating system, so that the required condenser pressure can be maintained.

When a float chamber in connection with the coil of an evaporator operated on the flooded system is used in this way, a supply of the liquid refrigerant should, in most cases, be by-passed around the float chamber so that only a small part of the total flow enters the chamber. This precaution is necessary because if all of the liquid refrigerant passed through the float chamber, there would be violent action comparable to boiling; and this sort of action would interefere with the proper operation of the float in regulating the level of the liquid refrigerant.

The float chamber is usually located so that it regulates the level of the liquid refrigerant in the accumulator of a flooded system rather than in the coil of the evaporator and for this reason is joined by equalizing connections both to the evaporator and to the accumulator.

Direct and Indirect (or Brine) Systems Compared.—In the *direct system* of refrigeration, the coil of the evaporator is placed in the storage rooms of the refrigerator, and liquid refrigerant is allowed to expand into them. In the *indirect* (or *brine*) *system*, the storage rooms are cooled by means of pipes filled with cold brine which has been previously cooled. Both systems have advantages and disadvantages. The indirect (or brine) system is more frequently used than the direct.

Indirect (or Brine) System.—A brine tank or a double-pipe *brine cooler* is required in the indirect system, and a pump must be used to keep the brine in circulation. The tank and the pump add to the *first cost* of the indirect system, making it greater than the cost of the direct system.

In small plants, the *operating cost* of the indirect system is less than that of the direct system; while, in very large plants, the direct system has the advantage. In order to maintain a fairly even temperature in the cold-storage compartments, the direct system must be operated night and day, as shutting down the plant means stopping the refrigeration process. On the other hand, if the quantity of the brine in the system is large, it may be kept in circulation by operating the brine pump during the night, while the rest of the plant is shut down. The pump is generally a small one of the centrifugal type. This type of pump requires little attention other than that which a night watchman

can give. In large plants, this method of plant operation is not satisfactory, because the quantity of brine is usually relatively small compared to the size of the plant.

In the indirect system, there are two transfers of heat. In the direct system, heat is removed directly from the substances to be refrigerated, and there is only one transfer of heat. In order to produce the same cooling effect with both systems, the liquid refrigerant must be evaporated at a lower temperature in the indirect system than in the direct system. This puts additional work upon the compressor, for an increase in the range of temperature and, consequently, also, in the range of pressure requires greater pressure limits for the operation of the compressor.

In general, a more even temperature of the compartments of the refrigerator may be obtained by the indirect than by the direct system. All systems are subjected to variations in temperature by (1) irregular flow of liquid refrigerant through the expansion valve and (2) variations of speed of the compressor. The temperature variations are readily taken up by the brine in the indirect system, as the brine has a high specific heat, and a large quantity of brine is affected. The direct system has the disadvantage that if a leak should occur, the vapor of nearly all refrigerants would injure many kinds of foods in the compartments of the refrigerator.

A larger quantity of refrigerant is necessary for charging a direct than an indirect system, and the refrigerant is expensive. On the other hand, the direct system requires, usually, a greater number of expansion valves than the indirect, as each bank of cooling coils in the direct system has an expansion valve.

In the indirect system, the brine-cooling coils, the brine pump, and the expansion valve can be located near the engine room, so that the parts which require frequent attention may be conveniently taken care of by the engineer. If the brine cooler is located in the engine room, it must be well insulated, or it will absorb large quantities of heat. This heat must then be removed, and considerably more work will be placed upon the refrigeration system. Neither system can be operated at the maximum efficiency when the compartments of the refrigerator are to be cooled to different temperatures.

Brine Cooler.—In the indirect or brine system of refrigeration, the brine after passing through the brine cooler is pumped through

pipes or coils in the storage compartments where refrigeration is needed. A brine cooler is often made like a double-pipe condenser, although larger pipes are used, the inner one being usually 2 inches in diameter, and the outer one 3 inches in diameter. The refrigerant passes between the pipes while the brine circulates in the smaller pipe. The countercurrent principle is utilized. In some brine coolers, the liquid refrigerant enters at the top, and the warm brine at the bottom, while, in others, the directions of flow are reversed. The advantages of a double-pipe brine cooler are the same as those of a double-pipe condenser.

CHAPTER III

PROPERTIES OF REFRIGERANTS

Properties of Refrigerants.—A substance which removes heat is called a *refrigerant.* The refrigerants most suitable for use in refrigerating systems are of two distinct kinds.

a. The first kind does not support combustion and will not produce an explosion. Refrigerants of this kind are *non-inflammable.*

b. The other kind will support combustion and may cause an explosion Such refrigerants are *inflammable.*

Some of the non-inflammable refrigerants are:

1. Carbon dioxide, CO_2.
2. Sulphur dioxide, SO_2.
3. Carbon tetrachloride, CCl_4.
4. Nitrous oxide, N_2O.
5. Dichlorodifluoromethane (sometimes called "F-12" or "K-12"), CCl_2F_2.
6. Carrene,* CH_2Cl_2.
7. Trieline, C_2HCl_3

The following refrigerants belong to a group of refrigerants that are to some extent inflammable:

1. Ammonia, NH_3.
2. Butane, C_4H_{10}.
3. Carbon bisulphide, CS_2.
4. Isobutane,† C_4H_{10}.
5. Methyl chloride, CH_3Cl.
6. Ethyl chloride, C_2H_5Cl.
7. Propane, C_3H_8.
8. Ethane, C_2H_6.
9. Chloroform, $CHCl_3$.
10. Ether, $(C_2H_5)_2O$.
11. Dieline, $C_2H_2Cl_2$.

* Carrene is generally classified as a non-inflammable refrigerant as the transportation companies do not require it to be marked with a shipping label to indicate inflammability.

† The chemical formula of isobutane is the same as that of butane

Boiling Temperatures of Commonly Used Refrigerants.—Approximate boiling temperatures at atmospheric pressure of the commonly used refrigerants are given in the following table:

	Degrees Fahrenheit
Ammonia	− 28
Sulphur dioxide	+ 14
Methyl chloride	− 10
Ethyl chloride	+ 55
Carbon dioxide (solid state)	−110
Dichlorodifluoromethane	− 22

Pressure Range.—The pressure in the cooling coils of the evaporator should preferably be somewhat higher than atmospheric, because, if the pressure is lower than atmospheric, air and moisture may enter the system through loose joints. Such air and moisture which enter by leakage are likely to cause trouble in the operation of the refrigerating device.

High pressures in a refrigerating system, and especially in the condenser, add greatly to the expense of construction, as extra-heavy pipes, fittings, stuffing boxes, etc., are required.

Cost of Replenishing Charge of Refrigerant.—Refrigerating systems might be made very simple if a suitable refrigerant were cheap enough to be discharged into the air after being used, in the same way that the exhaust gases from an automobile engine are discharged after combustion. All the refrigerants suitable for commercial purposes are too expensive to be discharged into the air after use and must, therefore, be converted to service over and over again. There is always some loss of refrigerant because of leakage through joints and stuffing boxes, and, after a time, impurities enter into it, making it unsuitable for further use, so that a new supply becomes necessary. For this reason, the cost of the refrigerant has some bearing on commercial applications.

Shipping Refrigerants.—Most refrigerants are shipped in containers called "cylinders" which have a pressure in them corresponding to the temperature of the refrigerant which is almost entirely in the liquid condition. These cylinders vary in size ranging from those intended for the transportation of 5 pounds of the refrigerant to larger ones intended for 200 pounds. Dieline, trieline, and carrene are usually shipped in steel barrels or drums containing about 300 pounds of the refrigerant.

Corrosive Action of Refrigerants.—Several refrigerants have a corrosive action upon some metals. For example, ammonia has a corrosive action on alloys of zinc and copper, especially brass. For this reason, brass or gun-metal fittings cannot be used in any part of a refrigerating system in which ammonia is the refrigerant. On the other hand, there are some refrigerants which do not attack metals appreciably when the refrigerant is in a pure state, but when it becomes mixed with foreign matter, such as oil, grease, or water, and especially if an emulsion is formed, there is likely to be considerable corrosion in spite of the fact that the pure refrigerant has no destructive effect.

Chemical Disintegration of Refrigerants.—A refrigerant which is to be circulated continuously in a refrigerating system must have a strong chemical bond to withstand repeated evaporations, condensations, and absorptions. In the compression system, some refrigerants have a tendency to disintegrate into their respective elementary substances, especially when working temperatures after compression are high. It should be noted here, however, that all the important refrigerants used in commercial refrigerating plants have chemical properties that permit them to be used without disintegration at the usual working pressures or temperatures.

Properties of Refrigerants as They Affect the Design of Refrigerating Systems.—The property of a refrigerant which has the most effect on the design of a refrigerating system is the pressure necessary to liquefy the vapor of the refrigerant in a condenser when using cooling water at ordinary temperatures. The volume of the gas or vapor of the refrigerant per pound at its pressure in the cooling coils of the evaporator is a determining factor in the size of the cylinder of the compressor in a compression system. Obviously, if this volume of the gas or vapor per pound is relatively small, the refrigerant is more suitable for commercial use than a refrigerant with a larger volume. On the other hand, the latent heat of evaporation of one pound of a really satisfactory refrigerant should be large, so that the minimum weight of the refrigerant may be circulated for a given refrigerating effect.[1]

[1] The *critical temperature* of the vapor of a refrigerant is the temperature above which the vapor cannot be condensed by the application of any pressure. It is obvious, of course, that the temperature of the gas or vapor in the condenser must be several degrees higher than that of the cooling

Ammonia.—The refrigerant which is most commonly used in large refrigerating plants is ammonia. It is not suitable, however, for use in the types of small refrigerating systems frequently used in apartment houses and dwellings. It has an offensive and penetrating odor which is very irritating to any of the animal membranes and especially to the eyes. The human body is not appreciably affected by an ammonia-and-air mixture of $\frac{1}{30}$ per cent during an exposure of 1 hour. This is the maximum amount of ammonia vapor that is not permanently injurious. The chemical symbol for ammonia is NH_3. In its natural state, it is a vapor which is very soluble in water. At atmospheric pressure, it boils at $-28°$ F. and has a critical temperature of 271° F.[1]

For reasons that will be explained later, only ammonia which is entirely free from water (*anhydrous*) can be used in a compression system of refrigeration.

It is desirable to use a refrigerant of which a small quantity can produce a large cooling effect. In order to have a small power equipment to keep the refrigerant in circulation through the system and to permit the use of a small cylinder in the compressor, the vapor of the refrigerant should have a small volume per pound; or, in other words, the weight of 1 cubic foot of the *vapor* should be large. This makes it possible to produce a given amount of refrigeration with a small compressor, as the size of the compressor and its power requirement depend on the volume of refrigerant circulating through the system. Ammonia possesses both of these desirable properties, and it is for this reason that it is so generally used in the large refrigerating plants.

Solubility and Purity.—Ammonia is very soluble in water, and at ordinary "room" temperatures, water will absorb about nine hundred times its own volume of ammonia vapor. At ordinary temperatures, it is also extremely volatile. It sometimes contains impurities[2] that reduce its effectiveness as a refrigerant. To test liquid ammonia for purity, pour a little

water circulating through the condenser. The critical temperature of the vapor, therefore, must be considered with respect to the maximum probable temperature of the cooling water.

[1] The temperature after compression in a compression type of refrigerating machine should not exceed about 300° F. because of the unstable condition of ammonia above this temperature.

[2] Ammonia should be free from pyridine, acetonitrile, naphthalene, hydrogen sulphide, organic acid, and other organic compounds.

into a test tube and allow it to evaporate; impurities will be left in the test tube.

Inflammability.—Ammonia gas or vapor at ordinary atmospheric temperatures does not burn readily in air, but when heated, it burns readily with a greenish-yellow flame. When it is heated to a higher temperature than 1600° F., it breaks up into its constituent gases (nitrogen and hydrogen), and, under some conditions, these gases form an explosive mixture,[1] especially when some oil vapor is present in a mixture of nitrogen, hydrogen, and ammonia vapor. An explosive mixture of this kind has been known to cause explosions of considerable violence in ammonia refrigerating plants. Ammonia should not be allowed to come into contact with a red-hot metal or an ordinary flame, for these will decompose the vapor of ammonia and ignite the resulting gases, thus causing an explosion.

Effect on Metals.—Ammonia, either as a liquid or as a vapor, does not react chemically on brass and similar alloys, but when the ammonia contains water vapor, ammonia hydroxide is formed, which has a very active chemical effect on copper, brass and, to a very limited extent, on steel. For all practical purposes, however, it may be said that iron and steel are not appreciably affected by exposure to ammonia. Some ammonia compressors having pistons made of copper do not show any appreciable deterioration after years of service.

Testing for Leaks.—Ammonia vapor is very light and readily finds its way through pipe joints. Special care should be taken in making all pipe connections. As ammonia has a disagreeable and pungent odor, a small leak is readily noticed, but its location is not so easily found. A water hose should always be in readiness to be used in case a large leak occurs. Water from the hose should be discharged upon the leak to absorb the ammonia and prevent serious damage. Sulphur candles[2] have proved very

[1] Explosive mixtures of ammonia vapor and air range from 13 per cent of ammonia and 87 per cent of air to 27 per cent of ammonia and 73 per cent of air.

[2] Sulphur sticks may easily be made by melting ordinary sulphur or brimstone in a tin can or ladle. The mixture should not get too hot, or it may burn. As an additional precaution to prevent burning, the tin can or ladle should be covered to exclude the air. Thin sticks of wood, about ¼ inch thick and 6 to 9 inches long, or strips of cardboard of the same size, may be dipped into the melted sulphur, the dipping being repeated until the coating on the sticks or strips is about 1/16 inch thick.

helpful in discovering leaks. The fumes given off by them when burning combine with the ammonia. If a burning sulphur candle is held near the nose, one may approach an ammonia leak without much danger. By using sulphur candles, a person who is caught in a compartment filled with ammonia gas can make his escape without serious injury. (For methods of testing water for ammonia leakage, see p. 221.)

Effect on Lubricants.—In the compression system of refrigeration, a considerable amount of lubricant is required in the com-

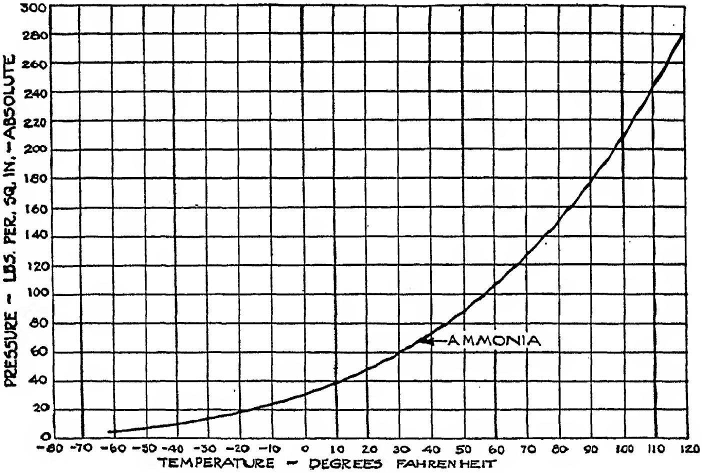


FIG. 51.—Pressure-temperature curve of ammonia.

pressor. The oil used for this lubrication obviously comes into contact with the ammonia. Anhydrous ammonia has little or no effect on petroleum lubricating oil. In the presence of moisture, however, at the usual temperatures in compressors, ammonia has the tendency to make an oil emulsion.[1]

In order to use the sulphur sticks for testing, the end of the sulphur coating should be lighted with a match, and the sulphur sticks should then be moved at a distance of about 1 inch, along all pipes, around all fittings, joints, valves, and stuffing boxes. As long as no gray smoke is noticeable, there is no leak, but as soon as smoke is observed while moving along some part of the piping, the place from which the smoke comes should be carefully investigated. The ammonia leak, no matter how small it is, will be found by this method.

[1] OSBORN, W. F., "The Physical Characteristics of Lubricating Oils as Applied to Refrigerating Machinery" in *Jour. Amer. Soc. Refrigerating Eng.*, Vol. 7, p. 166.

Charts and Physical Properties.—A comparison of the physical properties of ammonia and some other refrigerants is shown in Table IX in Appendix (p. 492). A pressure-temperature curve of ammonia is shown in Fig. 51. A so-called Mollier diagram giving the total heat content of ammonia for various conditions of pressure and percentage of vapor is given in the Appendix (facing p. 512). When ammonia is used as the refrigerant in a refrigerating system, only moderate pressures and temperatures are required.

Sulphur Dioxide.—When sulphur is burned, it combines with the oxygen in the air to form a vapor called *sulphur dioxide* (chemical symbol SO_2). Even in small concentrations in air, sulphur dioxide is very unpleasant to breathe, but it is not considered poisonous because its very presence is so suffocating that one will immediately seek fresh air. It is a colorless vapor, with the aforementioned pungent and suffocating odor. This vapor is very stable, and because it will easily withstand the temperature conditions of ordinary household refrigeration, it is used as a refrigerant in many of the small refrigerating machines intended for household use. Its boiling point at atmospheric pressure is 14° F., and its critical temperature is 311° F.

Inflammability.—Sulphur dioxide is not combustible, so that there is no danger of explosion.

Effect on Metals.—Pure sulphur dioxide has no corrosive effect on copper, copper alloys, zinc, iron, or steel unless water is in contact with the vapor, when sulphurous acid is formed which has some corrosive effect on copper, zinc, and iron. Because of this tendency of sulphur dioxide to form sulphurous acid, the presence of water vapor is detrimental and should not exceed 0.3 per cent by volume.

Testing for Leaks.—An easily applied method of locating leaks of sulphur dioxide is to apply ammonia water (aqua ammonia) to pipes and joints in which leaks are suspected. Sulphur dioxide in the presence of ammonia in any form becomes a dense white vapor.

Effect on Lubricants.—Sulphur dioxide has an objectionable effect on some kinds of petroleum oils commonly used for lubrication. To a certain extent, this refrigerant is self-lubricating; but, on the other hand, it has a tendency to absorb oils. The kinds of oil that have a light color are not so readily absorbed by sulphur dioxide as are dark oils.

Because of the tendency of sulphur dioxide to form sulphurous acid in the presence of even small quantities of water, it is necessary to be extremely careful in selecting oils that they contain no water.

Relative Displacement Compared with Ammonia.—A refrigerating machine using sulphur dioxide for the refrigerant requires about three times as much displacement in the cylinder of the compressor as a machine using ammonia for equal amounts of refrigeration.

Charts.—Figure 52 shows a pressure-temperature curve of sulphur dioxide, also similar curves for a number of other refrigerants.

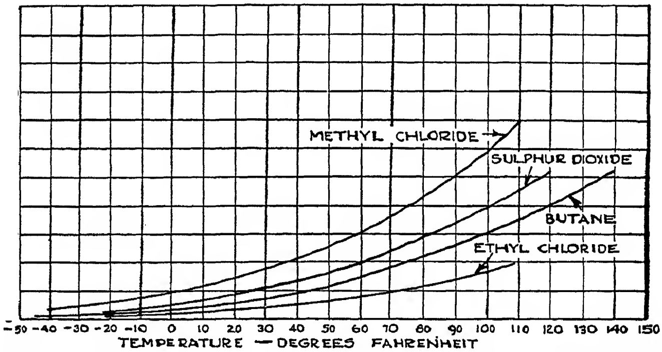


FIG. 52.—Pressure-temperature curves of sulphur dioxide, methyl chloride, butane and ethyl chloride.

A Mollier diagram giving the total heat content of sulphur dioxide for various conditions of pressure and percentage of vapor is given in the Appendix (facing p. 515).

Methyl Chloride.—By the chemical action of hydrochloric acid on methyl alcohol, a colorless, sweet-smelling vapor is obtained called *methyl chloride* (chemical symbol CH_3Cl). This vapor is easily liquefied when compressed and then cooled. Its odor resembles that of chloroform. Small quantities of methyl chloride when mixed with air have very little effect upon the lungs and eyes. When more than 10 per cent by volume is in the air, it acts as an anæsthetic, and larger concentrations of the vapor may produce death by suffocation. Because of this tendency to produce unconsciousness, it is not so safe as sulphur dioxide which is more irritating and consequently produces a constant urge to obtain pure air.

When methyl chloride is used as the refrigerant in a refrigerating system it is essential to dry out thoroughly the system before charging. One volume of water will take up three volumes of methyl chloride at ordinary temperatures and atmospheric pressure. The presence of water in methyl chloride will be noted by the formation of ice at the expansion valve. It is claimed by the manufacturers of a grade of methyl chloride known as "*Artic*" that it will not thus form ice when in contact with water under the conditions found in refrigerating systems.

The color of methyl chloride as well as its effects are somewhat like chloroform. It is very stable at the usual temperatures of household refrigeration and is now being used as a refrigerant in a number of household refrigerating systems. Its boiling point at atmospheric pressure is $-10°$ F.

Inflammability.—Methyl chloride is inflammable but does not burn readily. It is explosive under certain conditions, as, for example, when a mixture of from 10 to 15 per cent of methyl-chloride vapor and from 85 to 90 per cent of air is exposed to a spark or to a wire which is at a white heat. Mixtures of smaller concentrations will not explode. The flashpoint is about $-10°$ F. Rubber should never be used as a gasket material for making tight joints for valves and pipes containing methyl chloride as rubber is soluble in this refrigerant both in its liquid and in its vapor states. Specially prepared asbestos or chemically pure lead is suitable for this purpose.

Effect on Metals.—Methyl chloride does not affect copper, copper alloys, iron, or steel.

Testing for Leaks.—A large leak of methyl chloride may be noticed by the peculiar odor resembling chloroform. The exact location of the leak can be determined by the use of an alcohol flame and knowing that methyl chloride gives a greenish tinge to the nearly colorless flame of an alcohol lamp.

One manufacturer has recently placed on the market methyl chloride which is mixed with a chemical which is intended to give warning when there is a leakage of that refrigerant in the system. A warning agent of this kind must not, of course, lose its effect for safety upon persons after they become somewhat accustomed to it. It must be an irritant whose effects would become more and more severe with increased exposure. Acrolein (CH_2; CH·CHO) is a distinctive and powerful warning agent

and becomes increasingly effective with length of exposure. It is an eye irritant and produces lachrymation (shedding tears). A 1 per cent mixture in methyl chloride gives a satisfactory warning of 0.05 to 0.1 per cent by volume of methyl chloride in air. Acrolein may be used in conjunction with the *leak-detecting*[1] agent methyl nitrite. Acrolein has the disadvantage of being exceedingly toxic. It is rated as one of the most poisonous of all substances.[2] To safeguard the public there should be legislation which will require the use of a warning agent in *methyl chloride* when used in a refrigerating system.

Effect on Lubricants.—Methyl chloride dissolves practically all kinds of oils, so it has been found by experience that glycerin and white mineral oils are the only lubricants for compressors using this refrigerant. Because glycerin is hygroscopic, precautions must be taken to avoid the presence of water. Since methyl chloride often contains as much as 10 per cent of water, great care should be taken to obtain it of sufficient purity. The moisture in glycerin has been known to freeze in the cooling or evaporating coils of the refrigerating system and prevent the circulation of the refrigerant.

Although mineral oils are soluble in methyl chloride in all proportions, there are many of them that are good lubricants. Investigation has shown that the viscosity of oil is lowered by the presence of methyl chloride but not sufficiently to impair appreciably the oil as a useful lubricant. Some oils that give good results as lubricants for methyl-chloride compressors conform to the following tests: Flash point 320 to 400° F.; cold test -10 to $-20°$ F.; low sulphur content (below 0.15 per cent); no

[1] Another method of testing for methyl chloride leaks is to observe the color change of a test solution used to moisten a piece of absorbent cotton. The test solution for this purpose contains 3 grams of alpha-naphthylamine and sulphanilic acid which may be purchased in a test vial already mixed. (Roessler and Hasslacher Chemical Company, New York.)

In the application of this method of testing, a piece of absorbent cotton is made just large enough to cover the joint that is to be tested. This piece of absorbent cotton should then be carefully moistened without dripping and applied to the joint where the leak is suspected, and it should be kept there about 3 minutes. After that time, if there is a leak, it will be indicated by a red spot on the moistened absorbent cotton. Rubber gloves should be used when handling the test solution to prevent objectionable stains and odors.

[2] "International Critical Tables," Vol. II, p. 318.

soapy matter; viscosity 150 to 310 seconds (Saybolt[1] at 100° F.). Oils which do not conform exactly to these specifications may be used if it has been found by experience that they are satisfactory; but mineral oils for this use should contain not more than a slight trace of sulphur and no soapy matter. It is stated that the more highly refined oils, usually the colorless oils, are more satisfactory for difficult operating conditions than the oils that are less highly refined.

Relative Displacement Compared with Ammonia.—The cylinder of a compressor using methyl chloride will have a smaller displacement than one using sulphur dioxide for the same amount of refrigeration but requires larger displacement than an ammonia compressor.

The danger to life from escaping methyl chloride into air has been carefully studied by the U. S. Bureau of Mines, where tests were made to determine the possibility of poisoning by food and water that were contaminated by methyl chloride. These tests were conducted upon dogs with the following conclusions.[2]

1. No apparent signs of poisoning were caused by the average daily ingestion on four consecutive days of 550 grams of ground raw beef or 200 cubic centimeters of milk that had been exposed 15 to 75 hours to 100 per cent methyl chloride vapor at 35° F.

2. No apparent symptoms of poisoning or changes in the hemoglobin and blood cells were caused by drinking water nearly saturated with methyl chloride on 115 days of a total period of 171 test days. Also, no formates were found in the urine. Autopsy and examination of frozen sections, however, revealed a moderate degree of intracellular fatty degeneration affecting the ascending, descending, and collecting tubules of the kidneys. Analysis showed the water to be 75 to 100 per cent saturated with an average methyl chloride content of 0.595 gram per 100 cubic centimeters of water. This was the only water given the animals on 6 days of each week of the test.

3. The taste of water saturated with methyl chloride at 86° F. is sharp, sweetish, and sickening when first taken into the mouth, followed almost immediately by a burning sensation. Persons would not drink more than a mouthful or two. It was frequently refused by the animals, even though they were deprived of other water.

Charts.—A pressure-temperature curve of methyl chloride is shown in Fig. 52.

[1] See MOYER, JAMES A., "Power Plant Testing," Chap. xxi, McGraw-Hill Book Company, Inc., New York.

[2] See *Public Health Reports*, Vol. 45, No. 19, May, 1930.

Ethyl Chloride.—The odor of ether is a typical property of ethyl chloride (chemical symbol C_2H_5Cl). In its liquid form it is colorless, very volatile, and has a sweetish taste. Its vapor is very stable at the usual atmospheric temperatures and, for this reason, is suitable for household refrigerating machines and has also found considerable application in the refrigerating machines used on shipboard. Small quantities of ethyl chloride when mixed with air have very little effect upon the lungs and eyes. When more than 10 per cent by volume is in the air, it acts as an anæsthetic, and larger concentrations of the vapor may produce death by suffocation. Because of the tendency of this refrigerant to produce unconsciousness, it is not so safe as sulphur dioxide which is more irritating.

Inflammability.—Under some conditions, ethyl chloride is inflammable. When it is mixed with certain definite proportions of air, it burns with a greenish flame. When a mixture of from 5 to 14 per cent of ethyl-chloride vapor and from 90 to 86 per cent of air by volume is exposed to a spark or to a wire at white heat, there will be explosive combustion. Below or above these percentages, there will be no explosive effects. The flashpoint is below $-18°$ F. It is stated that by the addition of a small amount of methyl bromide to ethyl chloride, the mixture becomes non-inflammable and also non-explosive in air. The boiling point of ethyl chloride at atmospheric pressure is 55° F.

Effect on Metals.—Ethyl chloride has no corrosive effect on copper, copper alloys, iron, or steel.

Testing for Leaks.—Leaks in a system of piping containing ethyl chloride are difficult to locate, since this refrigerant, when expanded following compression, is below atmospheric pressure, and the tendency, therefore, is for air to enter the piping through leaks rather than for ethyl chloride to escape into the air.

Effect on Lubricants.—Ethyl chloride dissolves the oils and greases which are available for lubricating purposes. Experience has shown that glycerin is the only practical lubricant to be used with ethyl chloride, and the same precautions must be taken to avoid moisture as when methyl chloride is used.

Relative Displacement Compared with Ammonia.—Because of the light weight of ethyl chloride per unit of volume, the cylinder of the compressor using it for the refrigerant will be very large in comparison with ammonia and sulphur-dioxide compressors. A compressor using ethyl chloride as a refrigerant requires more

than twice the cylinder displacement of one using sulphur dioxide and about seven times the displacement of one using ammonia, for the same amount of refrigerating effect.

Charts.—A temperature-pressure curve of ethyl chloride is shown in Fig. 52. It is possible to operate the compressor of a refrigerating system using ethyl chloride at lower pressures than are commonly used with other refrigerants. Methyl chloride, for example, requires considerably higher operating pressures than ethyl chloride, as will be seen from a comparison of the curves in the figure.

Carbon Dioxide.—The gas which gives the "sparkle" to ginger ale and other carbonated liquids is carbon dioxide (chemical symbol CO_2). This gas is probably more generally known as the one resulting from the combustion of fuels such as coal, wood, oil, etc. It is a heavy, colorless, and odorless gas. It is found in small amounts in atmospheric air. It is harmless to breathe except in extremely large concentrations; it is harmful then because of the lack of oxygen in the air. This gas is very stable and will withstand the temperature conditions in ordinary household service. When carbon dioxide is subjected to a pressure of little more than 500 pounds per square inch and the temperature is about 32° F., it solidifies into a snowlike substance.

Because carbon dioxide is not irritating to the membranes of the body and is not dangerous as a poison, it is often preferred for use in refrigerating systems in hospitals and ships. If a person breathes air containing 2 per cent by volume of carbon dioxide for any length of time, however, his efficiency will be reduced, and headache and drowsiness will result. If very large quantities are taken into the lungs, death may be produced by suffocation.

Inflammability.—Carbon dioxide is not inflammable and has actually a tendency to smother any kind of combustion.

Effect on Metals.—Carbon dioxide has no corrosive effect on copper, copper alloys, iron, or steel.

Testing for Leaks.—Leaks of carbon-dioxide gas are difficult to locate, for the reason that the gas is colorless and odorless under practically all ordinary conditions.

Effect on Lubricants.—Carbon dioxide has no effect upon oils and greases. When, however, it is used as the refrigerant, there will be very low operating temperatures, and, for this reason, the oil must be suitable for low-temperature service. Glycerin

is usually preferred as a lubricant for carbon-dioxide refrigerating machines, as it does not freeze at the low temperatures that are required and is not affected by acid.[1]

Relative Displacement Compared with Ammonia.—Because of the large weight per unit of volume of carbon dioxide, its refrigerating effect for a given volume is also large. A refrigerating machine using carbon dioxide requires only about one-fifth as much cylinder displacement as a machine using ammonia for the same amount of refrigerating effect.

Charts.—The pressure-temperature curve for carbon dioxide is shown in Fig. 53. The compressor-discharge pressures range

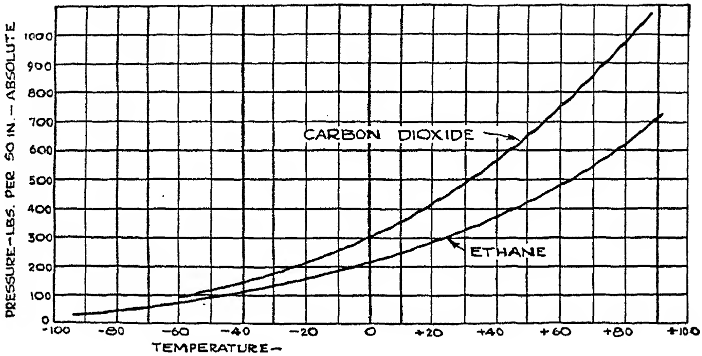


FIG. 53.—Pressure-temperature curves of carbon dioxide and ethane.

from 800 to 1,000 pounds per square inch with a corresponding range of suction pressures from 200 to 300 pounds per square inch. Because of the high discharge pressures which are necessary, the compressor, the condenser, and the piping connecting them must be designed for very great strength. A Mollier diagram of the properties of carbon dioxide is given in the Appendix (facing p. 515).

Dichlorodifluoromethane or "F-12" (CCl_2F_2).[2]—This refrigerant approaches nearly to the ideal refrigerant and was studied by F. Swartz. Later, it was developed by Dr. Thomas Midgley, the originator of ethyl gasoline.

[1] Carbon-dioxide gas absorbs water to form a carbonic acid (chemical symbol H_2CO_3). It does not have so much affinity for water as ammonia.

[2] The chemical reaction involved in the manufacture of dichlorodifluoromethane is as follows:

$$3CCl_4 + 2SbF_3 \xrightarrow[SbCl_5]{} 3CCl_2F_2 + 2SbCl_3.$$

When pure and dry antimony trifluoride (SbF_3) obtained by sublimation of the crude material is brought into contact with carbon tetrachloride (CCl_4) in the presence of a small amount of antimony pentachloride ($SbCl_5$), fluorine substitutes for chlorine in the carbon tetrachloride; since, fluorine substitution lowers the boiling point of the resulting compound by about 126° F.

A study of substances that have been used quite commonly as refrigerants shows that no refrigerant combines non-toxicity and non-inflammability with suitable refrigerating properties as does dichlorodifluoromethane.

A study made by the U. S. Bureau of Mines[1] shows that dichlorodifluoromethane may be regarded as practically non-toxic. Animals can withstand for long periods of time a concentration of some 20 per cent by volume.

The boiling point at atmospheric pressure is −21.7° F. It freezes at a temperature of −311° F. The latent heat of evaporation at atmospheric pressure is about 72 B.t.u. per pound. Dichlorodifluoromethane is only slightly soluble in water.

Inflammability.—Dichlorodifluoromethane is non-inflammable and non-explosive. Its fire-extinguishing ability may be judged by the fact that mixtures of 70 per cent of this refrigerant and 30 per cent butane by volume are still non-inflammable. Dichlorodifluoromethane is thermally stable up to 1000° F. It decomposes when heated by a flame in the presence of oxygen and water vapor, forming hydrochloric acid, hydrofluoric acid, carbon dioxide, chlorine and phosgene.

Effect on Metals.—In general, dichlorodifluoromethane is non-corrosive. The following metals have been tested: aluminum, cast iron, babbitt metal (67 per cent lead), monel metal, high- and low-carbon steel, all of which did not corrode in the presence of *dry* refrigerant at 235° F. for a period of 5 months. A few metals, like copper, bronze, and lead were slightly darkened but not corroded.

Dichlorodifluoromethane, *saturated with water* at room temperature, within 4 months at 235° F., corroded only Y-metal and magnesium alloy (94 per cent magnesium and 6 per cent aluminum); brass, copper and lead were discolored but not corroded. The solubility of this refrigerant in water at room

[1] *Report* of *Investigation*, 3013.

temperatures is small; hence, the equivalent amount of reaction is negligible.

The addition of the highly refined white mineral oils to dichlorodifluoromethane has no effect on corrosion. When dichlorodifluoromethane is passed slowly through a heated glass tube it breaks down at 1050° F; however, at 1400° F. complete decomposition occurs. In the presence of various metals, decomposition starts at different temperatures as follows: solder, 400° F; tin, 450° F.; lead, 620° F.; copper, 780° F.; and zinc, 786° F. It is extremely important to take the usual precautions of using dry tubing and thoroughly cleaned parts to avoid the possibility of getting any moisture into the system, as this water will form

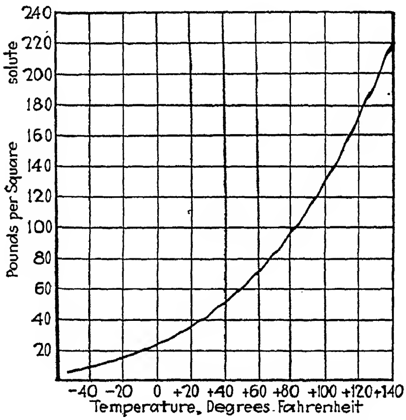


FIG. 54.—Pressure-temperature curve of dichlorodifluoromethane (F — 12).

ice at the needle valve in a flooded unit (p. 69) and interrupt the operation of the system.

Testing for Leaks.—By the use of a Halide lamp, the presence of a very small quantity of dichlorodifluoromethane may be detected. This lamp is filled with alcohol. A copper wire is heated by the lamp, and when a volatile substance containing chlorine comes into contact with the heated copper wire, the flame burns with a blue-green color. As dichlorodifluoromethane tends to remain around the leak or drop downward and does not diffuse rapidly, leaks may be easily detected by the use of this lamp.

Effect on Lubricants.—Dichlorodifluoromethane mixes with mineral oils in all proportions; consequently, there is no separation or oil blanket formed in the evaporator.

Relative Displacement Compared to Ammonia.—Because of the low heat content of dichlorodifluoromethane, the weight of refrigerant circulated per minute per ton of refrigeration is much higher than for other common refrigerants and is but a little higher than carbon dioxide. However, as the specific volume is much lower than for the other common refrigerants but higher than carbon dioxide, the theoretical piston displacement for dichlorodifluorormethane is about 1.69 times as great as for ammonia. In the case of sulphur dioxide the piston displacement is only about 62.6 per cent, and for methyl chloride it is only about 85 per cent as large.

Charts.—A pressure-temperature curve for dichlorodifluoromethane is shown in Fig. 54, and physical properties may be found from Table VI on page 513 in the Appendix.

Methylene Chloride (Carrene).—This refrigerant has been extensively used by the Carrier Engineering Corporation with centrifugal compressors. It is also known as dichloromethane.

Methylene chloride is a low-boiling chlorohydrocarbon having the chemical formula CH_2Cl_2. It is a colorless water-white liquid at ordinary atmospheric conditions. Methylene chloride has a sweet, pleasant odor quite similar to chloroform. At atmospheric pressure it boils at 103.64° F. The latent heat of evaporation at atmospheric pressure is 135.8 B.t.u. per pound. The critical temperature is 473° F.

Inflammability and Stability.—It does not form explosive mixtures with air. Pure oxygen or oxygen from the air has only a slight effect on the decomposition of methylene chloride up to 248° F. At higher temperatures it breaks down and forms phosgene ($COCl_2$), chlorine (Cl_2), and hydrochloric acid (HCl). The vapor of methylene chloride will extinguish the flame of a lighted taper.

A small quantity of water does not effect the stability of methylene chloride for temperatures below 212° F. The presence of a large quantity of water causes no decomposition of methylene chloride at its boiling point (atmospheric pressure). However, at still higher temperatures, some decomposition takes place.

Effect on Metals.—In general, soft steel, copper, aluminum, lead, and tin have no effect on the decomposition of methylene chloride. For temperatures up to 176° F., brass has a negligible effect. The combined effect of water and methylene chloride

on the above metals does not affect the decomposition of methylene chloride up to 212° F.

Relative Displacement.—At standard conditions (p. 106), about 1.485 pounds of methylene chloride must be circulated per minute per ton of refrigeration, which is equivalent to about 74 cubic feet per minute. This is about 21.5 times as great as for ammonia.

Charts.—The pressure-temperature curve for methylene chloride is shown in Fig. 55. The discharge pressure at 86° F. is about 20.48 inches of mercury absolute, while the suction pressure at 5° F. is about 2.39 inches of mercury absolute. The compression ratio is then about 8.56. For thermodynamic data see Table I.

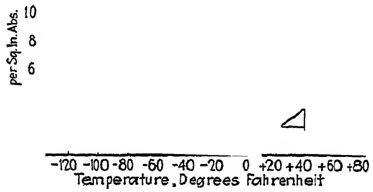


FIG. 55.—Pressure-temperature curve of methylene chloride.

Dichloroethylene.—Dichloroethylene is a low-boiling chlorohydrocarbon having the chemical formula $C_2H_2Cl_2$ or CHClCHCl. It is a colorless liquid having a specific gravity of 1.27 (water = 1). The boiling point at atmospheric pressure is about 122° F. The freezing point is −70° F. This refrigerant is also used with centrifugal compressors.

Inflammability and Stability.—Dichloroethylene is classified as an inflammable refrigerant. It does not burn readily as the flame propagates above the surface from which the refrigerant is evaporating and is quite readily extinguished. Dichloroethylene decomposes after 100 hours of heating at 680° F. into carbon and hydrochloric acid. It is an active liquid with a strong chloroform odor, while its vapor smells sweet.

It cannot be heated with alkalis as the treatment causes the liberation of the spontaneously explosive substance, chloroacetylene. Dichloroethylene dissolves cellulose acetate, rubber, oils, shellac, waxes, and resins.

The explosion limits are widened by the addition of dichloroethylene to mixtures of carbon monoxide and air, and this compound may become dangerous in some industrial applications.

Explosion limits of dichloroethylene are 3.5 per cent and 15 per cent by volume at 15° F. and atmospheric pressure.

Dichloroethylene inhaled by rabbits or injected under the skin causes deep breathing, narcosis, and finally death. Tests made on these animals showed abnormal amounts of fat in the liver and kidney.

Effect on Metals.—Dichloroethylene produces no reaction upon mild steel, wrought iron, nickel, copper, aluminum, and lead, when heated. However, with copper and aluminum, there is a slight discoloration. In the presence of water, dichloroethylene attacks metals.

Relative Displacement.—At standard conditions about 1.768 pounds of dichloroethlyene must be circulated per minute per ton of refrigeration. The volume of vapor per minute is about 108.4 cubic feet per ton of refrigeration. This is about 31.4 times as great as for ammonia.

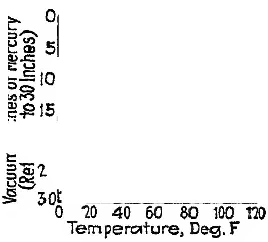


FIG. 56.—Pressure-temperature curve of dichloroethylene.

Charts.—The pressure-temperature curve for dichloroethylene is shown in Fig. 56. The discharge pressure at 86° F. is about 14.65 inches of mercury absolute, while the suction pressure is 1.78 inches of mercury absolute. The compression ratio is about 8.23. Other thermodynamic data may be found in Table I (p. 93, see footnote).

Trichloroethylene (Trieline).—Trichloroethylene is another refrigerant used with centrifugal compressors. It is also a low-boiling chlorohydrocarbon having the chemical formula C_2HCl_3. It is a heavy, colorless, mobile liquid having a pleasant odor. Trichloroethylene boils at atmospheric pressure at 188° F. The latent heat of evaporation at atmospheric pressure is 104.5 B.t.u. per pound. The freezing point is −126° F.

Inflammability and Stability.—Trichlorethylene is non-inflammable and non-explosive; in fact, it may be used to extinguish fires. The fire hazard is considered small as it is in the class with that of chloroform. Trichloroethylene is used for dry cleaning in place of other inflammable cleaners. At ordinary temperatures and conditions it is stable.

The presence of oxygen will cause some decomposition when trichloroethylene is exposed to sunlight, but in the dark this is

negligible. This is quite true of all chlorohydrocarbons. Water does not dissolve readily in trichloroethylene, and therefore it can be stored in open tanks under a covering of water which floats on top and acts as a seal. In general though, closed tanks are advised in order to keep out dust, dirt, and to prevent evaporation.

Effect on Metals.—Moisture and metals have no apparent effect on the stability of trichloroethylene. At temperatures even above the boiling point of the solvent with excess water, trichloroethylene does not hydrolyze to any great extent. Metals such as tin, aluminum, copper, iron, lead, and steel do not increase decomposition of dry or water-saturated trichloroethylene at ordinary temperatures and pressures.

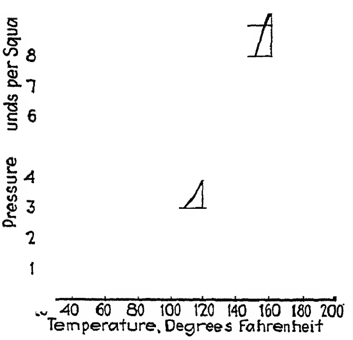


FIG. 57.—Pressure-temperature curve of trichloroethylene.

Relative Displacement.—At standard conditions about 2.173 pounds of trichloroethylene must be circulated per minute per ton of refrigeration. This is equivalent to about 513 cubic feet per minute per ton of refrigeration, which is much higher than the above refrigerants but is lower than water.

Charts.—The pressure-temperature curve for trichloroethylene is shown in Fig. 57. The discharge pressure at 86° F. is 3.42 inches of mercury absolute, and the evaporator pressure at 5° F. is 0.315 inch of mercury absolute. Other data are given in Table I.

Ethane (C_2H_6) and Propane (C_3H_8).—Very little is known regarding the physiological properties of ethane; and the similar properties of propane are known only by inference from its effects on rats, as studied and reported by Dr. E. E. Smith as follows:

> When the quantity of propane gas is about 5 per cent and is inhaled for an hour, slight drowsiness is produced, but when the quantity is moderately large (37.5 to 51.7 per cent) and is inhaled for two hours, muscular weakness is observed which is followed by mild anæsthesia. When the quantity of gas is as much as 70 per cent and is inhaled for several hours, there will be muscular spasms followed by anæsthesia.

The temperature-pressure curve of ethane is shown in Fig. 53. It will be observed that the curve for ethane approaches the pressure range of carbon dioxide but is not quite so high. The curve for propane is similar to the pressure-temperature curve of ammonia.

Butane and Isobutane (C_4H_{10}).—The physiological properties of butane are probably similar to those produced by propane, with a possibility that butane is more toxic. The physiological properties of isobutane are probably about the same as those of butane.

The pressure-temperature curve of butane is shown in Fig. 52. It will be noticed that this curve is somewhat similar to the pressure-temperature curve of ethyl chloride.

Ether.—Refrigerating machines of the piston type using ether as the refrigerant are bulky and cannot be used to produce intense

TABLE I

	Dieline[1] $C_2H_2Cl_2$		Trieline[1] C_2HCl_3		Carrene[1] CH_2Cl_2	
	"Standard-ton" temperatures					
	5° F.	86° F.	5° F.	86° F.	5° F.	86° F.
Absolute pressure, pounds per square inch	0.82		0.16	1.72	1.29	10.3
Volume of liquid, cubic feet per pound	0.0127		0.0109		0.012	
Volume of vapor, cubic feet per pound	63.00	8.5	240.00	25.20	48.3	6.8
Latent heat of evaporation, B.t.u. per pound	136.00	133.00	112.50	109.50	149.00	146.00
Entropy of liquid	0.0029	0 0425	0.0025	0.0368	0.0037	0.0535
Entropy of evaporation	0.2930	0 2435	0.2420	0.2010	0.3200	0.2670
Entropy of vapor	0.2959	0 2860	0.2445	0.2378	0.323	0.3205
Specific heat of liquid	0.270		0.233		0.340	
Specific heat of vapor (constant pressure)	0.1625		0.120		0.154	
Specific heat of vapor (constant volume)	0.1425		0.105		0.131	
Specific gravity of liquid (water = 1)	1.27		1.47		1.33	
Specific gravity of vapor (air = 1)	3.36		4.55		3.00	
Critical absolute pressure, pounds per square inch	800[2]				1,490[2]	
Critical temperature, degrees Fahrenheit	470[2]				380[1]	

[1] The refrigerant dieline is stable *dichloroethylene*, trieline is trichloroethylene, and carrene is a water-white liquid at normal atmospheric conditions. It has a slightly sweetish odor similar to dieline but is non-combustible. Carrene is *methylene chloride*.

[2] Approximate.

cold, because the absolute pressure of ether vapor is only about 1.3 pounds per square inch at 4° F., and to make it evaporate at any temperature nearly as low as this would require an excessively large cylinder of the compressor. This large size would not only make the piston type of compressor clumsy and costly but would also involve much waste of power in mechanical friction. The tendency of air to leak into the system is another practical objection to the use of a pressure so low.

Heating and Cooling.—Refrigeration means the removal of heat from a substance in order to produce a low temperature. When heat is added to a substance, its temperature is raised; and when it is removed, its temperature is lowered. The removal of heat is sometimes called the *production of cold,* as though *cold* had a different meaning from *heat.* *Hot* and *cold* are terms which apply to effects produced upon our senses and indicate merely relative temperatures.

When coal burns in a stove, it gives heat to the stove, which, in turn, gives it up to the cooking pans and their contents. A spoon left in one of these pans becomes hot or absorbs heat until it and the contents of the pan have the same temperature. If a pan is taken from a hot stove and is placed in a basin of cold water, it loses some of its heat to the water and continues to do so until the water has the same temperature as the pan and its contents. A freshly killed fowl hung in a refrigerating compartment gives off heat and raises the temperature of the surrounding air. The air, in turn, has its heat removed by cold brine (indirect refrigeration, p. 70) or by ammonia vapor (direct refrigeration), which circulates in the cooling or evaporating coils.

Measurement of Temperature.—If one's hand touches a piece of ice, the sensation of cold is noticed. If a live coal is touched, the sensation of heat is observed. Temperature, then, is the sensation which these bodies produce upon the senses or, more exactly, upon instruments used to measure the intensity of heat. The temperature of a substance may not show directly the quantity of heat in that substance. For example, if a pail contains 10 pounds of water at 60° F., and another contains 2 pounds of water at 150° F., the first pail, although at a lower temperature, has the greater quantity of heat. Temperature, therefore, reveals only the state or intensity of heat, not the amount of heat in a body.

If two bodies have the same temperature, there is no transfer of heat from one to the other. But if one body is at a higher

temperature than the other, heat will pass from the higher to the lower temperature.

If 122° F. is to be changed into centigrade degrees, it is first necessary to subtract from it 32 and then multiply this result (122 − 32 = 90) by $^{100}/_{180}$ or $^{5}/_{9}$, and the result is 50° C. Using a short cut, the above methods of calculation may be stated as follows:

$$\text{Centigrade degrees} = \tfrac{5}{9} \times (\text{Fahrenheit degrees} - 32°)$$
$$\text{Fahrenheit degrees} = \tfrac{9}{5} \times (\text{Centigrade degrees}) + 32°$$

Measurement of Heat. Heat Units.—The most common way of measuring heat is to observe its effect in raising the temperature of a quantity of water. The quantity of heat is then determined from the rise in temperature of the water and its weight. The unit of heat is the *British thermal unit,* generally abbreviated B.t.u. The B.t.u. is the amount of heat required to raise the temperature of 1 pound of water 1° F. A more recent way of defining this heat unit is to state its value as $^{1}/_{180}$ of the heat required to raise the temperature of 1 pound of water from 32 to 212° F. at normal atmospheric pressure. This is generally called the *mean B.t.u.* The unit may easily be remembered by the phrase "1 pound of water, 1° F."

Latent Heat of Fusion.—The addition of heat to a substance may produce other changes besides one in temperature. The form or condition of the substance may be altered, as when iron melts or water boils. While this change of condition is taking place, there is no change in temperature, yet large quantities of heat are absorbed. If heat is applied to ice, its temperature rises to 32° F. When this point is reached, a further addition of heat does not change the temperature until all of the ice is melted. The amount of heat added to produce this change in condition at atmospheric pressure has been found by experiment to be 144 B.t.u. (see p. 106) for each pound of ice which is melted. The heat which is thus expended is called the *latent heat of fusion* of ice. Because of its capacity to absorb large quantities of heat in melting and because of its relative cheapness, ice is used extensively in reducing the temperature of warm substances, when, of course, the temperature need not be lower than 32° F. When ice is melted in a refrigerator and is changed to the liquid state as water, a great deal of heat is absorbed from the contents of the refrigerator, and this absorbed heat is carried away through

the drain to the outside. Unless ice in the refrigerator melts,[1] it will not cool the contents of the refrigerator and the enclosed air.

Heat of Liquid.—If water is heated in an open vessel over a fire, its temperature will rise until it reaches 212° F. Since the water absorbs heat and there is *no change in its condition,* the heat which is absorbed is said to be the *heat of the liquid.* To each pound of water, then, a certain number of B.t.u. is added, depending, of course, on the amount of heat already in the water. Since water freezes at 32° F., at this temperature it is said to have *no heat of the liquid.* If the temperature of 1 pound of water is raised from 32 to 212° F., the water absorbs 212 − 32 or 180 B.t.u., which is the heat of the liquid at 212° F.

Latent Heat of Evaporation.—Evaporation is changing a liquid to a vapor or a gas by the application of heat. The heat which is added to a liquid to bring it from 32° F. to the temperature of boiling is, of course, the heat of the liquid. When, however, the boiling point is reached, the temperature remains constant, and a relatively enormous amount of heat must be added to change the liquid to a vapor or a gas. This amount of heat added during the change from the liquid to the vapor or gas state is called the *latent heat of evaporation,* and although, *in principle,* this latent heat is exactly similar to the latent heat of fusion which is absorbed by a solid, as, for example, ice in melting, it is a very much larger quantity of heat in all practical cases which have to do with refrigeration.[2]

Evaporation takes place from the surface of liquids at all ordinary temperatures, but when heat is applied, it will take place more rapidly and at a higher and higher rate as the temperature is increased, until, finally, it takes place not only on the surface but also all through the body of the liquid at minute surfaces

[1] Except in emergencies, it is not a good plan to put newspapers over the ice in a refrigerator, nor is it good policy to put so-called "blankets" over ice in order to prevent its melting.

[2] According to the molecular theory, during evaporation some molecules of a refrigerant, or, in fact, any liquid, find their way outside the surface of the liquid, and these "outside" molecules tend to fill the surrounding space; if this space is enclosed, there will be impact on the walls with a resulting pressure which is called the *vapor pressure* of the liquid. This amount of pressure has a definite relation to the temperature. When the temperature of a liquid is obtainable, it is possible to determine the vapor pressure of the liquid by reference to tables of its properties.

which are formed by little bubbles. When a liquid is boiling, there is a very much greater surface for evaporation, largely because of the formation of bubbles, than when there is only slow evaporation at the surface, and, consequently, the liquid will change to a vapor at a higher rate in proportion to this greater area of the effective surface for evaporation. These bubbles of vapor will form and collect into larger bubbles as they rise to the surface.[1]

When a liquid evaporates, work must be done upon it (1) to separate the molecules against their attractive forces, (2) to make space for the newly formed vapor by pressure against the surrounding medium, doing work against its pressure. The work of the first kind is called *internal latent heat;* the second, *external latent heat.* The sum of these two is the *total latent heat* given in the tables of the properties of refrigerants.

The *latent heat of evaporation* of a refrigerant may readily be given up by changing its vapor back to the liquid condition. To evaporate 1 pound of *liquid* ammonia at a temperature of 5° F. and a pressure of 34.3 pounds per square inch absolute, requires the addition of 565 B.t.u.; and to change 1 pound of ammonia *vapor* which is at a temperature of 5° F. into liquid ammonia, there must be removed from it 565 B.t.u.

The evaporation of any liquid causes a cooling effect. The more rapid the evaporation the greater the effect of cooling. Volatile liquids like gasoline, ether, and alcohol, which have large latent heats of evaporation, can be used for cooling. In refrigerating plants, cooling is produced by the evaporation of liquid ammonia, liquid carbon dioxide, or some other refrigerant.

Specific Heat.—In practical refrigeration work, it is necessary to consider not only the amount of heat absorbed in cooling different substances but also the amount of heat absorbed by the walls within which the substances are stored.

In a preceding paragraph, the B.t.u. was defined as the amount of heat necessary to raise the temperature of 1 pound of water 1° F. It has also been shown that all substances do not require the same amount of heat to raise their temperature 1° F. For example, if a pound of water and a pound of lead are heated over

[1] Before the bubbles that are formed within the liquid can collect and rise toward the surface of the liquid, the vapor pressure within the bubbles must be sufficient to overcome the pressure on them due to the weight of the liquid over them.

the same fire, the lead will have the higher temperature. This means that the quantity of heat which will raise 1 pound of water to a certain temperature will raise 1 pound of lead to a much higher temperature. Then, too, if these two substances have the same temperature and they are cooled to the same lower temperature, in doing so the water will give up more heat than the lead. In order to know how much heat is given off by a certain substance in cooling, it is necessary to know the specific heat of the substance. The specific heat of a substance is the amount of heat required to raise or lower the temperature of 1 pound of the substance 1° F. The specific heat of water is taken as the "point of departure" or the standard, and the specific heat of every other substance is determined by the relation of the amount of heat required to raise or lower the temperature of 1 pound 1° F. to 1, which is the adopted specific heat of water. Thus, the specific heat of alcohol is 0.60; of iron, 0.12, of ice, 0.50. To illustrate concretely, since the specific heat of ice is 0.5, the amount of heat which is necessary to raise the temperature of a given weight of ice 1° F. will raise the temperature of the same weight of water 0.5° F.

Example.—How many British heat units are required to lower the temperature of 500 pounds of aluminum from 100 to 40° F., the specific heat of aluminum being 0.22?

As it requires the removal of 0.22 B.t.u. to lower the temperature of 1 pound of aluminum 1° F. it will require the removal of $0.22 \times 500 \times (100 - 40)$ or 6,600 B.t.u. to lower its temperature from 100 to 40° F.

Example.—A storage room 50 by 25 by 10 feet is to have the air cooled from 70 to 32° F. At the lower (final) temperature, a cubic foot of air weighs 0.0807 pound and the specific heat at constant pressure is 0.237. How much heat must be removed from the air?

The room contains $50 \times 25 \times 10$ or 12,500 cubic feet of air. Since 1 cubic foot of air at 32° F. weighs 0.0807 pound, the weight of air in the storage room when cooled is $12{,}500 \times 0.0807$ or 1,008.75 pounds. To cool each pound of air, at constant pressure, it is necessary to remove 0.237 B.t.u. for each degree Fahrenheit change in temperature. To cool 1,008.75 pounds of air from 70 to 32° F. requires $1{,}008.75 \times 0.237 \times (70 - 32)$ B.t.u. or 9,084.8 B.t.u. to be absorbed.

If this amount of heat is to be removed by means of melting ice, it will require $9{,}084.8 \div 144$ or 63.1 pounds, since each pound in melting absorbs 144 B.t.u.

Heat Involved in Refrigeration Process.—When foods and other goods are placed in cold storage, there is first a reduction in temperature. To calculate the amount of heat to be removed, it is necessary to know the weight of the foods and other goods,

their specific heats, and the required change in temperature. The sum of the products of these quantities for each kind of goods in cold storage gives the amount of heat which must be removed.

It so happens that some foods are usually kept frozen in cold storage. In this case, it is necessary to know the latent heat of fusion of the foods which are to be frozen, because this latent heat must be removed when actual freezing takes place. As the largest portion of all foods is water, they will freeze at 28 to 32° F., and in freezing they require the removal of a large quantity of heat. Foods are sometimes cooled below 32° F. In such cases, in order to find the amount of heat to be removed, it is

Table II.—Heat Properties of Various Food Substances

Substance	Specific heat before freezing, B.t.u. per pound, per degree Fahrenheit	Latent heat, B.t.u. per pound	Specific heat after freezing, B.t.u. per pound, per degree Fahrenheit	Temperature at which usually kept, degrees Fahrenheit
Apples	0.92	122		30
Asparagus		135		34
Bacon		30		30–32
Bananas		108		35–40
Beef	0.68	102	0.38	33–35
Beets	....	129		
Butter	0.64	15	0.37	0–15
Cantaloupes		128	....	33–36
Carrots	0.87	124	0.45	30–36
Cheese	0.64	46	0.37	28–34
Cream	0.68	84	0.38	34
Eggs	0.76	92	0.40	31–33
Fish	0.82	109	0.43	20
Ice cream	0.78	90–105	0.45	−10–5
Milk		124	0.47	34
Mince meat		56		
Mutton	0.81	96	0.67	32
Oranges	....	124	....	32–35
Oysters	0.84	114	0.44	30–35
Parsnips		119		32–35
Peaches		125		30
Pears		120		30–32
Pork	0.51	66	0.30	40
Poultry	0.80	93	0.42	20–28
Strawberries	....	130		33–40
Veal	0.70	90	0.39	36

necessary to know the *specific heat* of the foods *before* they freeze and also *after* they are frozen. When the specific heat, the latent heat, the weight, and the final temperature are known, it will be possible to find the amount of heat (in B.t.u.) that must be removed.

Example.—How much heat must be removed from 5 tons of poultry in cooling it from a temperature of 65 to 20° F.?

The amount of heat which is to be removed will be considered in three parts: (1) heat to be extracted in lowering the temperature from 65 to 32° F.; (2) latent heat at 32° F.; (3) heat removed in lowering the temperature from 32 to 20° F. In general, it is customary to find these three quantities for 1 pound of the substance which is to be cooled.

	B.t.u.
1. Heat removed per pound from 65 to 32° F. = 0.80(65 − 32)	26.4
2. Latent heat per pound at 32° F.	93.0
3. Heat removed per pound from 32 to 20° F. = 0.42(32 − 20)	5.04
Total heat removed per pound	124.44

Total heat removed for 5 tons = 5 × 2,000 × 124.44 = 1,244,400 B.t.u.

Recently, Prof. Willis R. Woolrich[1] has suggested the use of the formula $h_f = 143.3\frac{w}{100}$ for calculating the latent heat of fusion (h_f) of foodstuffs containing w per cent of water.

Theoretical Displacement of Compressor.—The theoretical displacement of the cylinder of a compressor in a refrigerating system is calculated by multiplying together the weight of refrigerant in pounds per minute, which is circulated through the cooling coils of the evaporator, and the volume in cubic feet per pound of the refrigerant at the temperature of the suction side of the compressor.

If, for example, the weight of ammonia circulated per minute in pounds per ton of refrigeration is 0.4 and the volume of 1 pound of ammonia vapor at the suction temperature of the compressor (5° F.) is 8.15 cubic feet, the theoretical displacement of the ammonia compressor, in this case, is, then, 8.15 × 0.4 or 3.26 cubic feet per minute. This is the theoretical volume (cubic feet) of saturated ammonia vapor which will enter the suction pipe of the compressor from the cooling coil of the evaporator per minute, in order to produce a ton of refrigeration for the required temperature conditions. The actual displacement of a

[1] "Latent Heats of Foodstuffs," *Refrigerating Eng.*, Vol. 22, No. 1, p. 21, July, 1931.

cylinder of a compressor must always be considerably larger than the theoretical value as calculated by this method, because the weight of ammonia vapor in the cylinder is always considerably less than the amount that is theoretically displaced by the piston. In other words, the actual displacement volume per ton of refrigeration must always be larger than the theoretical amount.

Superheating of Vapor Due to Compression.—The pressure on the suction side of a refrigerating system is the pressure in the cooling coils of the evaporator. The compressor takes the vapor from the evaporator and compresses it to a higher pressure. In doing this, a certain amount of work is done upon the vapor and its temperature is raised above the boiling point at the higher pressure. Vapor under these conditions is said to be *superheated.* If this additional heat is left in the vapor, the condenser will have to do more cooling than if there were no superheat.

Wet and Dry Compression of Ammonia.—There are two systems by which ammonia compressors are operated: (1) dry compression and (2) wet compression. If the ammonia vapor which is drawn into the cylinder of a compressor does not contain particles of liquid ammonia in suspension, the compression is said to be *dry;* and, conversely, if the vapor taken in by the compressor does contain particles of liquid ammonia in suspension, the compression is said to be *wet.*

With dry compression, the vapor of the refrigerant is superheated during compression, because work is being done on the vapor. The water jacket of the compressor reduces only slightly the amount of superheat and does not prevent entirely the superheating of the vapor of the refrigerant.

In a *wet-compression* ammonia refrigerating system, the heat produced by the compression of the ammonia vapor raises its temperature and causes the evaporation of the liquid ammonia which was injected. In evaporating, the latent heat of evaporation of the liquid ammonia is absorbed by the superheated compressed vapor, and its temperature is lowered. The evaporation of even a little liquid ammonia prevents superheating.

In the ammonia refrigerating system in Fig. 13, the liquid ammonia used for this cooling purpose is taken from the liquid receiver as shown by the piping.

Another means often used to prevent superheating is to inject cold oil into the cylinder at the beginning of the compression

stroke. This oil absorbs the superheat. After leaving the compressor, the oil is removed from the compressed vapor by the *oil separator*. Oil injected in this way serves to seal the piston and valves, thus preventing leakage. It also partly fills the clearance space at the end of the stroke and causes the expulsion of nearly all of the vapor; this leaves the cylinder to take in nearly a full charge on the return stroke.

There is a difference of opinion as to which of these systems is the better. The presence of liquid refrigerant in the cylinder may result in disaster to the machine if the clearance space is small. The valves should be designed to permit the escape of any of the excess of the liquid on the compression stroke.

Comparative Value of Refrigerants.—In order to compare refrigerants, a practical range of temperature has been chosen, the standard temperature of the gas or vapor *after compression* being 86° F., and the lower temperature, that is, the temperature in the cooling coil of the evaporator, being 5° F. Table III shows the important properties of ammonia and carbon dioxide for these two temperatures.

Item 3 in Table IV, (p. 104), shows that the *evaporation* of 1 pound of *ammonia* will produce a cooling effect of 474.4 B.t.u. per pound, while 1 pound of *carbon dioxide* in evaporating will produce a cooling effect of only 54.2 B.t.u. per pound. In order, therefore, to produce the same amount of refrigeration as is derived from 1 pound of ammonia, there must be 474.4 ÷ 54.2 or 8.75 pounds of carbon dioxide in circulation. In this respect, ammonia has a great advantage over carbon dioxide.

TABLE III

	Temperature, degrees Fahrenheit	Absolute pressure, pounds per square inch	Latent heat of evaporation, B.t.u. per pound	Specific volume, cubic feet per pound
Ammonia	86	169.2	492.6	1.77
	5	34.3	565.0	8.15
Carbon dioxide	86	1,039.0	27.0	0.47
	5	334.2	114.7	0.27

In order to produce a temperature of 86° F. at the end of compression, the absolute pressure of carbon dioxide will be 1,039 pounds per square inch; and for ammonia, 169.2 pounds per

square inch. The pressure required when using carbon dioxide is 1,039.0 ÷ 169.2 or 6.04 times as great as that for ammonia. Such a high pressure requires a compressor and piping of heavy construction. When carbon dioxide is used as the refrigerant, pressures as high as 1,100 pounds per square inch absolute are frequent in tropical countries, as it is difficult to obtain a supply of cooling water with a sufficiently low temperature to permit having a discharge pressure much under this value. Pressures in ammonia refrigerating systems are neither very high nor very low but are above atmospheric pressure; hence, no special construction of the compressor or piping is necessary.

In producing a given cooling effect with carbon dioxide, it is necessary to circulate (as calculated in the preceding paragraph) 8.75 times as much carbon dioxide as would be required to obtain the same cooling effect with ammonia. Now, since the volume of 1 pound of carbon dioxide at 5° F. is 0.27 cubic feet, 8.75 times as much, or 2.36 cubic feet of carbon-dioxide gas, must be removed by the compressor as compared to 8.15 cubic feet of ammonia vapor (see table) for the same cooling effect.

The principal characteristic properties of five of the refrigerants most commonly used are given in Table IV (p. 104). The comparison is made on the basis of the number of pounds of refrigerant that must be circulated in the system per minute to produce one ton of refrigeration.[1]

A comparison is also made of the theoretical displacement of the compressor per ton of refrigeration and the theoretical horsepower required per ton of refrigeration. The values given in the table are based on a temperature of 5° F. in the cooling coil of the evaporator and a temperature of 86° F. in that portion of the condenser where the gas or vapor is in the saturated condition.

The number of pounds of refrigerant which it is necessary to evaporate per minute per ton of refrigeration is given in item 7 in the table, and the theoretical displacement of the compressor per minute per ton of refrigeration is stated in item 9. To obtain the *actual displacement* of a compressor from the values given in item 9, it would be necessary to increase the values given from 15 to 25 per cent. The *theoretical horsepower required per ton of refrigeration* (see p. 277) is shown in item 10, where a comparison of values shows that the horsepower per ton of refrigeration does not vary a great deal for four of the refrigerants, these being

[1] For definition of 1 ton of refrigeration, see p. 106.

TABLE IV.—CHARACTERISTICS OF REFRIGERANTS
For Standard Temperatures of 86° F. in Condenser and 5° F. in Evaporator

	Ammonia	Carbon dioxide	Sulphur dioxide	Ethyl chloride	Methyl chloride	Dichlorodifluoromethane ("F-12")	Dieline	Trieline	Carrene
1. Absolute pressure at 5° F.	34.27	331.8	11.82	4.65	20.89	26.51	1.78*	0.315*	2.39*
2. Absolute pressure at 86° F.	169.2	1,039.6	65.9	27.1	95.53	107.9	14.65*	3.42*	20.48*
3. Ratio of compression	4.93	3.14	5.57	5.83	4.57	4.07	8.23	10.84	8.56
4. Total heat content of vapor leaving evaporator, B.t.u. per pound	613.3	101	162.3	165	154	78.79	137.35	113.67	163.80
5. Heat of liquid, leaving condenser, B.t.u. per pound	138.9	46.8	18.2	23	25.1	27.72	23.22	20.1	29.24
6. Refrigerating effect, B.t.u. per pound	474.4	54.2	144.1	142	138.9	51.07	114.13	93.57	134.56
7. Pounds of refrigerant per minute per ton refrigeration	0.42	3.7	1.40	1.41	1.45	3.92	1.75	2.137	1.48
8. Specific volume of vapor in evaporator, cubic feet per pound	8.15	0.27	6.66	17.1	4.72	1.485	63	240	48.3
9. Theoretical displacement per minute per ton of refrigeration, cubic feet	3.44	0.99	9.26	23.95	6.83	5.82	110	513	71.5
10. Horsepower per ton of refrig.	0.99	1.87	0.99	0.92	1.06	1.00	0.918	0.928	0.965
11. Coefficient of performance	4.77	2.52	4.77	5.13	4.45	4.72	5.14	5.09	4.9

* Pressures for dieline, trieline, and carrene are in inches of mercury.

ammonia, sulphur dioxide, methyl chloride, and ethyl chloride. On the other hand, the horsepower per ton of refrigeration required when carbon dioxide is used as the refrigerant is double the power needed for any one of the four other refrigerants in the table.

For the usual temperatures in a refrigerating system, ammonia is compressed to only relatively moderate pressures, and the necessary displacement of the compressor is also relatively small, as shown by item 9. The advantages of ammonia for use as the refrigerant in large refrigerating plants so much outweigh the disadvantages that it is used much more than any other refrigerant.

Evaporation and Condensation of Refrigerants.—Water and ammonia and other liquids act very much alike with reference to their boiling points and evaporation. At normal atmospheric pressure, water boils and becomes steam or vapor at a temperature of 212° F. Liquid ammonia boils and becomes a vapor at about 28° F. below zero (−28° F.). At more than atmospheric pressure, water must be raised to a greater temperature than 212° F. in order to boil. For instance, if the gage pressure is increased to 50 pounds per square inch, water will not boil or vaporize until its temperature is 298° F. If ammonia is at a pressure which is greater than that of the atmosphere, it will not become a vapor at −28° F. but at some higher temperature, according to the pressure to which it is subjected. For each pressure there is a definite temperature at which liquid ammonia or water will boil and vaporize. If liquid ammonia is heated at its boiling point, some of the liquid will evaporate and become a vapor and in doing so will take up its latent heat of evaporation. Then, if this ammonia vapor has its temperature lowered below the boiling point, it will give up its latent heat and condense to the liquid condition. Ammonia and other refrigerants take the condition of a vapor or a liquid according to whether the actual temperature is above or below the boiling temperature corresponding to the pressure. For example, if ammonia at a gage pressure of 28.4 pounds per square inch has its temperature raised above 15° F., it will be a vapor; if its temperature is below 15° F., it will be a liquid. Likewise, if ammonia is at a gage pressure of 181.1 pounds per square inch and has its temperature raised above 95° F., it is a vapor, and below 95° F. it is a liquid.

It follows then that increasing the pressure of a refrigerant raises the boiling and the condensing temperatures, and, con-

versely, decreasing the pressure lowers the boiling and the condensing temperatures. Any vapor, moreover, condenses at constant pressure when its temperature is lowered below the boiling point.

Refrigeration Units.—Refrigeration capacity is usually measured in terms of the standard commercial *ton of refrigeration per day*, meaning the quantity of heat in B.t.u. required to melt 1 ton of pure ice at 32° F. into water at 32° F. in 24 hours. The latent heat of fusion of ice is 144 B.t.u.[1] per pound at this temperature. A standard commercial ton of refrigeration per day is, therefore, a quantity of heat equal to 2,000 × 144 or 288,000 B.t.u. per 24 hours, which is sometimes called *refrigeration power*. Refrigeration capacity is simply calculated by dividing the total heat transfer in a day by 288,000. If, for example, a refrigerating plant transfers 3,000,000 B.t.u. in 24 hours, the refrigeration capacity of the plant is 3,000,000 divided by 288,000 or 10.41 tons of refrigeration per day.

The capacity of a refrigerating machine is usually expressed in tons of refrigeration or ice-making effect in 24 hours.

The joint committee of the American Society of Mechanical Engineers and the American Society of Refrigerating Engineers on Standard Tonnage Basis of Refrigeration recommends the following units:

a. A standard ton of refrigeration is 288,000 B.t.u.

This value is obtained by multiplying together 2,000 pounds (1 ton) and 144, which is the latent heat of fusion of ice at 32° F. in B.t.u. per pound.

b. The *standard commercial ton of refrigeration capacity* is the rate of cooling at 200 B.t.u. per minute.

c. The standard rating of a refrigerating system using liquefiable gas or vapor is the number of *standard* tons of refrigeration it performs under adopted pressures of refrigerants, namely: (1) the inlet (suction) pressure being that which corresponds to a saturation temperature of 5° F. (−15° C.) and the discharge pressure being that which corresponds to a saturation temperature of 86° F. (30° C.).

The following equivalents of a standard *ton of refrigeration per 24 hours* are convenient for reference:

[1] The U. S. Bureau of Standards has made a more exact determination in which it was found that the latent heat or fusion of ice was almost exactly 143.5. This figure may, however, be considered as so near to the usually accepted value, which is 144, that the latter number continues to be used as the standard value for all practical calculations.

B.t.u.
288,000 per day
12,000 per hour
200 per minute

Ice-making Capacity.—Although the standard for the usual commercial purposes is the *ton of refrigeration per* 24 *hours*, there is also another term which is sometimes used in the rating of refrigerating machines, especially in plants where the machines are used for making ice. This other unit is called *ice-making capacity* and means the number of tons of ice which a refrigerating machine can produce in 24 hours. This quantity is usually equal to about 50 to 75 per cent of the refrigerating capacity as expressed in tons of refrigeration per day. The heat transfer necessary to produce ice includes the following four items: (1) heat removal to cool the water to the freezing point; (2) heat removal to freeze the water at the constant temperature of 32° F. (3) heat removal to cool the ice to the temperature of the "brine bath"; (4) heat removal to make up losses.

In order to make these items clearer, the following example is given: Water at a temperature of 90° F. is to be used to make ice by the use of brine which has been cooled to 15° F. The first item of cooling as outlined in the preceding tabulation, therefore, is to cool the water from 90 to 32° F. This is a temperature change of 58°. It will be sufficiently accurate to assume that the specific heat of water is 1.0. Now, the difference in temperature times the specific heat of water is the quantity of heat change per pound of water which is cooled, and this quantity of heat is 58 B.t.u. It requires the absorption of 144 B.t.u. to melt 1 pound of ice, and the specific heat of ice is approximately 0.5, so that the amount of heat required to cool the ice to the temperature of the brine is equal to the temperature range multiplied by 0.5. The difference between 32 and 15° F. is 17° F., and the heat change necessary for this cooling is 17×0.5 or 8.5 B.t.u. per pound of ice. The total heat required, therefore, to *cool the water* to the freezing point to *make ice* and then to *cool the ice* to the temperature of the brine is $58 + 144 + 8.5$ or 210.5 B.t.u. per pound. At least 10 per cent additional heat units must be added to make up the loss of heat through the insulation of the brine tank or coils and other incidental losses, so that the total quantity of heat which is required for ice making is $210.5 + (210.5 \times 0.10)$ or nearly 232 B.t.u. per pound of ice. This last

figure is equivalent, of course, to 2,000 × 232 or 464,000 B.t.u. per *ton of ice* which is made.

The relative amount of refrigeration required to produce a ton of ice may be found by dividing the actual refrigeration required to produce a ton of ice by the heat removed to produce a standard ton of refrigeration. In the above case there is removed 464,000 B.t.u. per ton of ice which, if divided by 288,000 B.t.u., will show that there are 1.61 tons of refrigeration required to produce a ton of ice; or 0.61 ton of refrigeration must be supplied to lower the temperature of the water, cool and ice below 32° F, and allow for losses.

Mechanical Equivalent of Heat.—It has already been shown that heat may be made to do work by the use of some form of heat engine. Since work and heat are interchangeable forms of energy there must be some relation between the B.t.u. and the foot-pound. By actual experiments, it has been found that 778 foot-pounds of work are equivalent to 1 B.t.u. This quantity 778 is called the *mechanical equivalent of heat.*

CHAPTER IV

COMPRESSORS FOR REFRIGERATING PLANTS

Classification of Refrigerating Compressors.—Refrigerating compressors may be classified as follows: (1) reciprocating piston, (2) gear, (3) rotary, (4) centrifugal. In the past the piston reciprocating compressor of the slow-speed type was commonly used, but with the development of other refrigerants than those commonly used in the past other types of compressors have been produced. In the case of piston compressors the ratio of length of stroke to diameter of cylinder varies from 1 to 1.25 with piston speeds as high as 750 to 800 feet per minute. The chief factor limiting the speed of the piston type of compressors is the difficulty in obtaining valve areas large enough with the low lift necessary for silent operation, combined with the proper speed of the entering and leaving vapor. Gear compressors (p. 129) have been developed to be used with methyl chloride and some other refrigerants. This type of compressor has been constructed by one manufacturer for capacities of 15 and 60 tons for commercial use, and gear compressors have also been used, to some extent, in household units. The centrifugal compressor (p. 131) has been used extensively abroad. This type of compressor is, of course, limited as to the pressures to be handled economically. The speeds are quite high, as well as the capacities.

Compressors for Ammonia Refrigerating Systems.—The compressor in an ammonia refrigerating system is usually of the reciprocating rather than the rotating type. The reciprocating kind has a piston which moves up and down or back and forth when the compressor is operated. Every compressor has a suction or intake stroke and a discharge or "exhaust" stroke. During the suction stroke, the vapor of the refrigerant is drawn into the cylinder filling it as the piston moves in the direction which increases the cylinder volume to be filled. During this suction stroke, the suction valves of the compressor are open, and the discharge valves are closed. When the piston gets to the end of its suction stroke and the normal amount of vapor which

the cylinder will hold has been drawn in, the suction valves close, the discharge valves remain closed, and the piston begins a stroke in a backward direction compared with the suction stroke. This "backward" stroke is called the *compression*, because, during this stroke, the vapor inside the cylinder is compressed into a smaller and smaller space until, near the end of the stroke, it occupies only a small part of the space it occupied before compression. The vapor is then at a high pressure and very hot. During the compression stroke, the pressure increases until it is slightly higher than that which is maintained in the *condenser;* and when this pressure is reached, at about three-quarters of the compression stroke, the discharge valves open, and the hot[1] compressed vapor is forced into the condenser.

Reciprocating Ammonia Compressors.—Reciprocating compressors are classified as (1) single-acting and (2) double-acting. A single-acting compressor takes in the vapor of the refrigerant to be compressed on only one side of the piston, so that only one charge is compressed in a revolution. This type of compressor does not require a stuffing box so intricate and expensive as the one in a double-acting compressor. A double-acting compressor takes in the vapor of the refrigerant on both sides of the piston so that two charges of vapor are compressed in a revolution. A double-acting compressor, therefore, has approximately twice the capacity of a single-acting compressor having the same diameter of cylinder and length of stroke.

A compressor consists in its most essential parts of (1) a cylinder in which a piston moves to and fro and (2) the suction and discharge valves in this cylinder. In most cases, these valves are operated by springs so that they open when the pressure is in the direction to lift them and close when this pressure is released.

The available data regarding recent installations of refrigerating systems seem to show that there are in use considerably more vertical, single-acting compressors than horizontal double-acting compressors. Each type has its advantages. The strongest point in favor of the selection of the horizontal type of machine is the accessibility of the working parts all of which are near the floor, compared with the location of the same parts in a large vertical compressor. The construction of a horizontal com-

[1] The vapor of the refrigerant becomes very hot because of the expenditure of mechanical energy or work in compressing it from the low pressure in the cooling coils of the evaporator to the high pressure required so that it will force itself into the condenser against the pressure prevailing there.

pressor makes a compact arrangement that is easily accessible for operating, overhauling, and repairing. This type of compressor can be operated efficiently when working under *either wet or dry compression* (see p. 101). It is a dependable machine for either kind of operation, while the vertical type, unless very carefully designed, may give trouble when it must be operated with wet vapor. The horizontal type of compressor, furthermore, requires only a little more floor space than the vertical type, while the headroom it requires is only about one-half that of a vertical machine.

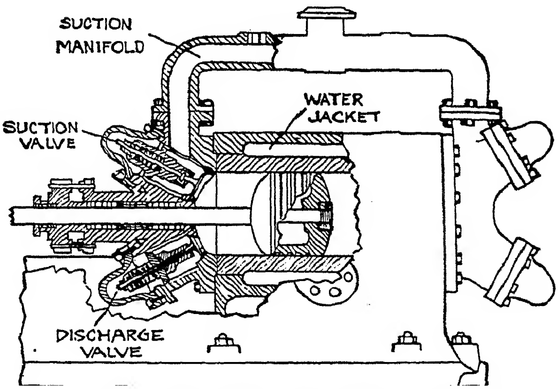


FIG. 58.—Horizontal double-acting compressor.

Piston of Compressor.—The piston of a horizontal double-acting compressor is generally a light and hollow semi-steel casting. The cylinders of many horizontal compressors have spherical heads, as in Fig. 58, and when this is the case, the faces of the piston must be of a shape similar to the ends of the cylinder. Such a piston consists of two parts, one part fitting against a tapered shoulder on the piston rod. A nut and locknut are then screwed onto the piston rod to fasten this half of the piston tightly against the shoulder. The other half of the piston is held in place by a locknut which completely fills a recess made for it in the piston. The pistons of compressors which operate at high speeds differ mainly from those of the medium-speed types in having flat instead of spherical heads. Vertical single-acting compressors usually have pistons with flat heads and suction valves in the piston, as shown in Fig. 59.

Stuffing Box of Horizontal Compressor.—A long and deep stuffing box must be provided to fit around the piston rod of a horizontal double-acting compressor, to prevent the leakage of the refrigerant. Provision must be made to keep the stuffing box cool so that the piston rod may slide back and forth through it with as little friction as possible and so as not to score or other-

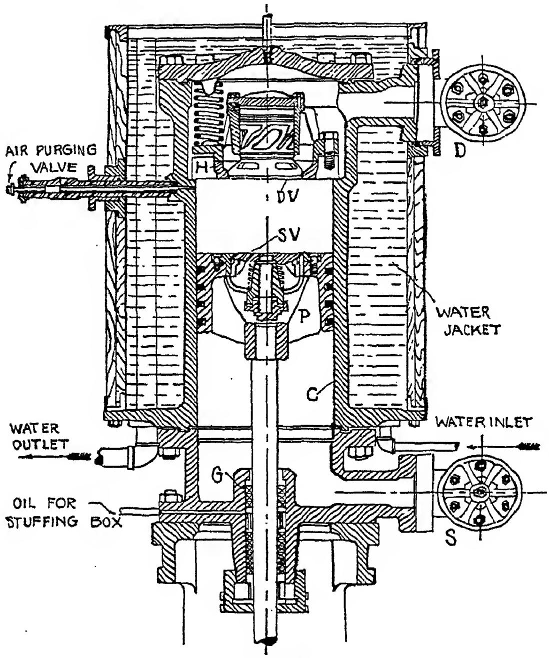


Fig. 59.—Cylinder of vertical single-acting compressor (uni-flow).

wise damage the rod. A well-designed stuffing box of a horizontal compressor contains two sets of packing separated by a space called the *lantern* which is filled with oil. The set of packing at the outer end of the stuffing box (farthest from the cylinder) is kept in place under pressure by means of the usual type of gland for a stuffing box. This packing is held tightly in place in the stuffing box by a nut which screws into an extension of the gland. There are a number of devices for pushing the gland of the stuffing box evenly into the space provided for the packing. Such stuffing boxes are usually provided with an *oil lantern* connected to oil piping which supplies a lubricant under pressure,

as at A, in Fig. 60. On the other side of the oil lantern is a pipe B connected to the suction manifold of the compressor (the operation of this compressor is explained on p. 112). Details of an oil lantern are shown more clearly in Fig. 61, and Fig. 62 shows an arrangement of two stuffing boxes as used in the compressor shown in Fig. 78.

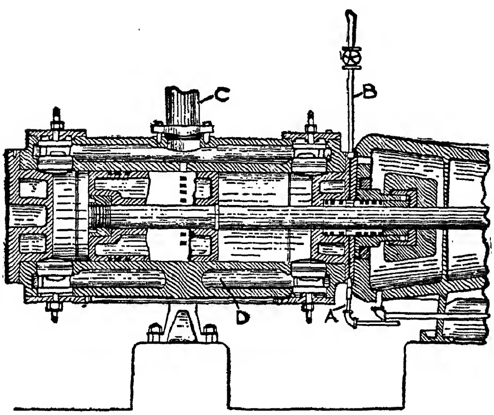


FIG. 60.—Compressor with openings in side wall of cylinder for supplementary filling (showing oil piping).

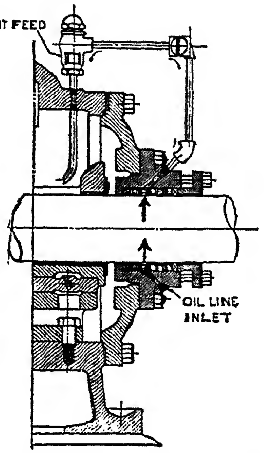


FIG. 61.—Sectional view of stuffing box for metallic packing and oil connections to lantern.

Valves for Compressors.—The valves for medium-speed horizontal compressors are generally of the balanced poppet type and are usually made of high-grade steel. Suction valves are made with enlarged sections of the stems to prevent their falling into the cylinder of the compressor if they should break; the suction valves in Fig. 58 have collars on the valve stems for this purpose. Figures 63 and 64 show details of the poppet type of valves used in some vertical compressors. The suction valves open and close by the difference between the pressure within the cylinder and the pressure in the suction pipe. Similarly, the discharge valves open and close by the difference in pressure in the cylinder and the discharge pipe. Springs are placed on the valve stems, which force the valves to seat and hold them in place. The suction valves open inward while the discharge valves open outward. At about three-quarters of the compres-

sion stroke, the vapor of the refrigerant, which is at high pressure, automatically opens the discharge valves and passes out of the cylinder. At the end of this compression stroke, the discharge valves close automatically by the action of the springs and prevent the compressed vapor from returning into the cylinder.

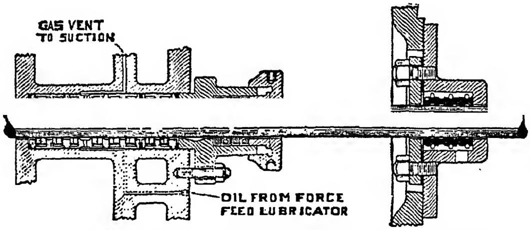


Fig. 62.—Details of metallic packing for ammonia compressor shown in Fig. 78.

In most of the high-speed horizontal compressors, the valves are placed radially around the ends of the cylinder. There is this important difference between the cylinder of a high-speed and of a medium-speed horizontal compressor. In a medium-speed machine, the valves are placed in the heads of the cylinder, while, in a high-speed machine, there are no valves or valve chambers in the cylinder heads, which are made, preferably, of semi-steel.

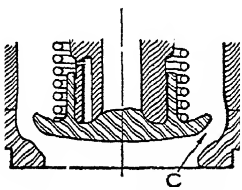


Fig. 63.—Compressor discharge valve of poppet type.

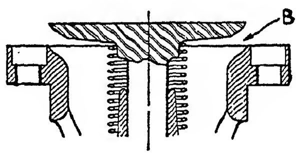


Fig. 64.—Compressor suction valve of poppet type.

The valves of a horizontal high-speed ammonia compressor are usually of the *ring-plate* or the *strip-plate* (*feather*) type and are made of unusually high-grade steel by special heat treatment, so that they may be thin and light. The *ring-plate* type of valve is held on its seat by means of a set of relatively light spiral springs *S*, *S*, as shown in Figs. 65 and 66. Another ring-type valve is shown in Figs. 67 and 68, which give detailed views of both suction and discharge valves. A compressor of this kind has a valve area as large as can be put in a given diameter of cylinder.

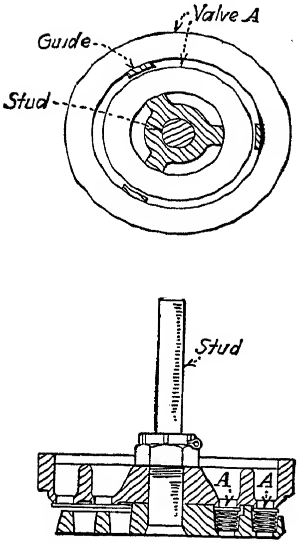


FIG. 65.—Ring-plate type of suction valve.

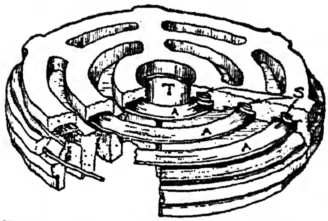


FIG. 66.—Details of typical ring-plate valve.

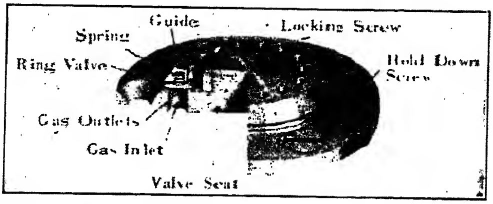


FIG. 67.—Single-ring suction valve.

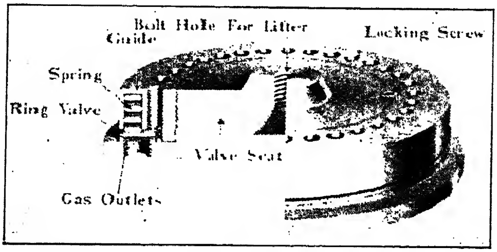


FIG. 68.—Single-ring discharge valve.

Very large valve area is not an advantage, however, unless all the valves can be made tight.

Strip-plate (*feather*) valves are made of thin steel plates placed over a grid *G* (Figs. 69 and 70) which has slots through which the vapor of the refrigerant flows. These thin strip-plate valves are held in place on the grid by the "back plate" or guide *P*, which is channeled out in the shape of an arch, the ends of which come close enough to the grid *G* to hold the valve against the face of the grid. The pressure of the vapor of the refrigerant must, therefore, bend the thin steel plate valves in order to make a passage for the vapor into or out of the cylinder. The detail at the right-hand side of the figure shows by arrows the flow of the vapor through the slots in the valve seat and then through the openings between the valve seats and the steel plates at the bottom of the valves. This type of valve requires no springs, as it is held against the seat by the shape of a "back plate" which holds down the ends of the valve, thus requiring it to bend or spring away from its seat when opening and in this way producing an opening between the valve and its seat through which the vapor of the refrigerant must pass. This passage for the vapor is in the shape of an arc of a circle conforming to the shape of the "backplate." The ends of the valve remain in contact with the seat when the valve is open for the flow of vapor through it, while the middle portion of the valve is forced up against the "back plate." This kind of valve is shown also in Figs. 71 and 72, with other details.

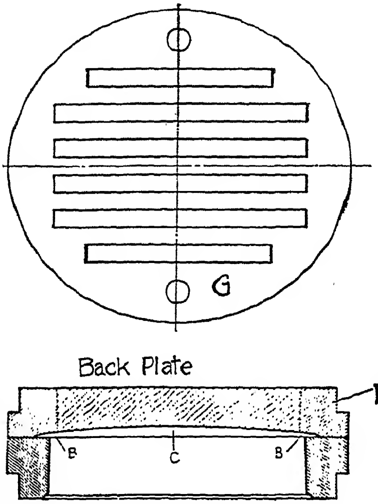


FIG. 69.—Typical strip-plate type of compressor valve.

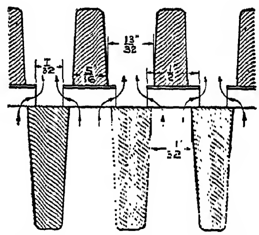


FIG. 70.—Details of strip-plate valve.

These two types of valves for high-speed horizontal compressors are designed to permit a very large opening through the valves with provision, at the same time, for noiseless operation. The disadvantage of the ring-plate type of valve is that, when it is used in a compressor, the clearance is considerably more than in

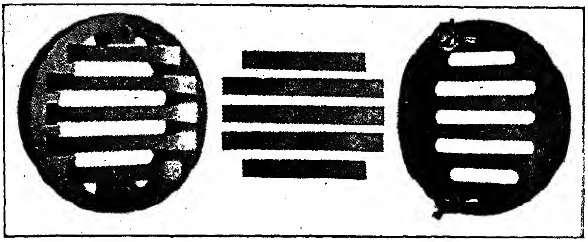


FIG. 71.—Valves of feather type.

compressors having the type of valves suitable for slow and medium-speed operation. A moderate amount of clearance does not increase the power required per ton of refrigeration so much as was once supposed. Clearance does reduce the capacity of a compressor, but if there is not more than 3 or 4 per cent of clearance, the horsepower required per ton of refrigeration will not be much different for a compressor with this amount of clearance than for one which has the smallest possible amount. When superheated vapor remains in the clearance space at the end of the compression stroke and expands during the following suction stroke, it does *work on the piston.* This work is, of course, nearly equal to the *work done on an equal weight of the vapor* during compression. On account of this expansion, however, the *effective* length of the stroke of the compressor is reduced; but the practical way out of this difficulty is to increase the length of the stroke when a new compressor is being designed or to increase the speed at which the compressor operates.

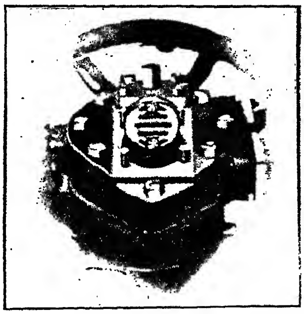


FIG. 72.—Portion of engine cylinder showing location of feather valves.

Water Jacket.—When the vapor of the refrigerant is compressed, the work done on the vapor increases its temperature and superheats it. Superheating the vapor does two things:

(1) It requires more heat removal by the *condenser;* and (2) it increases the volume of the compressed vapor; this, in turn, increases the work of the *compressor.* In order to reduce somewhat these effects, the cylinders of many compressors are constructed with a *water jacket,* the purpose of the water jacket being to cool the cylinder and thus to reduce the amount of superheating. A reduction in superheat means less work for both the condenser and the compressor.

A water jacket is not effective for removing all the superheat, because the temperature of the vapor of the refrigerant must be raised above the temperature of the cooling water before the water can begin to remove heat, and the temperature of the vapor does not rise above the temperature of the water in the jacket until a considerable portion of the stroke is completed. Also, the flow of heat from the vapor to the dry metal surfaces of the cylinder is slow, so that only a small amount of heat developed during the compression stroke is absorbed by the water circulating in the jacket.

Lubrication of High-speed Compressors.—As in the case of all high-speed machinery, an adequate system of lubrication is extremely necessary for the bearing surfaces. It is a common practice in modern refrigerating plants to use an automatic central oiling system consisting of an oil filter, storage tank, water separator, oil pump, and sight feed oil indicators. By the use of such a system, the oil can be used over and over again.

Vertical Single-acting Ammonia Compressors.—The tendency in designing ammonia compressors is toward higher operating speeds; and within recent years, it has been possible to obtain compressors with capacities from 25 to 50 tons of refrigeration per day, operating at speeds from 400 to 500 revolutions per minute. The advantages of these high speeds are obvious. Electric power is being used more and more for refrigerating plants, and it is desirable to avoid speed reduction from the motor to the compressor in order to obtain an efficient drive. Other advantages are (1) saving in space required; and (2) weight, which means lower first cost.

Vertical single-acting compressors are generally provided with a false head *H* (Fig. 59) sometimes called a *safety head.* This false head permits the cylinder to operate with a smaller clearance than would otherwise be safe. The safety head is movable but is held in place by stiff spiral springs. In case liquid refriger-

ant or broken parts enter the cylinder, the false head is pushed upward, thus preventing serious injury to the cylinder. During the next downward stroke, the false head moves back into place. The discharge valves are placed in the safety head. In case these fail to open, the pressure in the cylinder is relieved by the movement of the safety head, which prevents the cylinder from being subjected to excessively high pressures. If, under the same circumstances, a compressor of this kind had a solid head, there would be serious breakage of the parts of the cylinder. Also, in the case of an overcharge of liquid refrigerant in a vertical compressor having a very small clearance, the safety head prevents serious damage, the only effect being a somewhat noisy hammering of the valves and the head because of the presence of this liquid in the cylinder.

A single-acting compressor is generally of a vertical type similar to the one shown in Fig. 59 in which the cylinder *C*, piston ***P***, stuffing box *G*, and other parts are marked. In this type of compressor, the vapor of the refrigerant enters the cylinder near the bottom through the suction pipe *S*. On the *upward* stroke, the cold vapor of the refrigerant, which comes from the cooling coils through the suction pipe, is drawn into the cylinder space ***below*** the piston. During the return (downward) stroke, the vapor passes through the suction valve *SV* (in the piston) into the space on the *top side* of the piston. At the end of the downward stroke, the suction valve closes, and the vapor of the refrigerant is then compressed during the next upward stroke. When the pressure of the vapor in the cylinder becomes great enough, the discharge valve opens, and the vapor is forced out of the cylinder.

By the method of making the surfaces of the top of the piston and the cylinder smooth and parallel, the clearance space can be made very small.

The cylinder is cooled by the circulation of water in a *water jacket*. The arrows in Fig. 59 indicate the direction of flow of the water through the water jacket. The bottom head of the cylinder is provided with a stuffing box. Oil is fed into a circular space in the stuffing box under pressure through an oil pipe, as shown in the figure. In the top of the cylinder head, there is a valve for *purging* the cylinder of air when necessary.

The compressor shown in Fig. 73 has two cylinders of the vertical single-acting, uniflow type. Suction and discharge valves are of the *automatic poppet* kind, shown in Figs. 63 and 64. Fig-

ure 74 shows the piping arrangements for the vertical compressor shown in Fig. 73.

Instead of single inlet and discharge valves, this compressor has a "nest," or several poppet valves, by which a larger valve area for a given size of cylinder is obtained, the object being to secure a quick and complete filling of the cylinder and a steady discharge pressure. For a very large capacity, several of these high-speed compressors are generally connected in parallel on the suction and discharge ammonia lines.

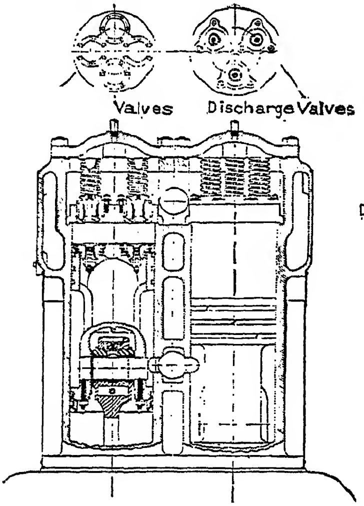


Fig. 73.—Vertical single-acting high-speed ammonia compressor.

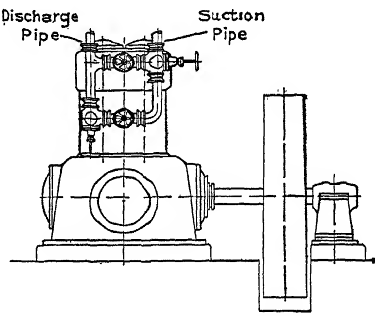
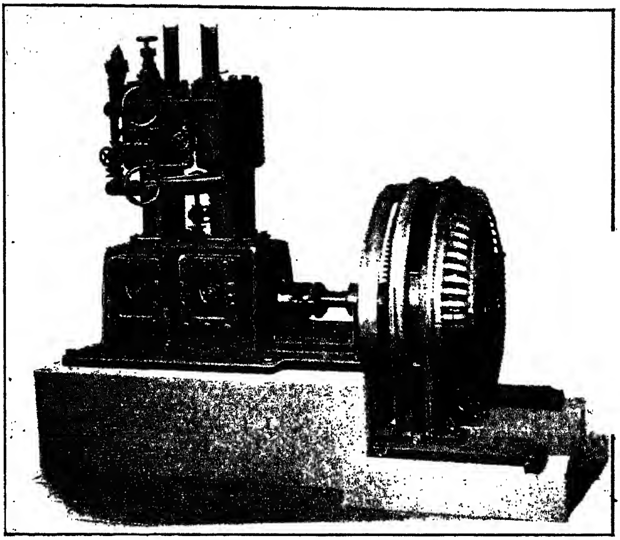


Fig. 74.—Pipe connections for vertical compressor.

A high-speed compressor also of the vertical single-acting type is shown in Fig. 75. It has a special large suction valve in the piston and a large plate discharge valve, the latter in the form of a thin ring covering an annular opening in the cylinder head. The valve arrangements of this compressor are shown in Figs. 67 and 68. The safety-head spring is marked *A* in Fig. 77, where the locations of discharge valve *D* and suction valve *S* are marked. Detailed views of the valves are given in Fig. 68. They are ring type, extremely light weight, giving full gas area with an extremely small lift, to obtain quiet operation at all speeds. The compressor may be either belt driven or directly connected to a synchronous electric motor, Diesel-oil engine, or steam engine. Figure 78 is a sectional drawing of an ammonia compressor showing a stuffing box for an enclosed crank case.



75.—Vertical high-speed compressor driven by synchronous electric motor.

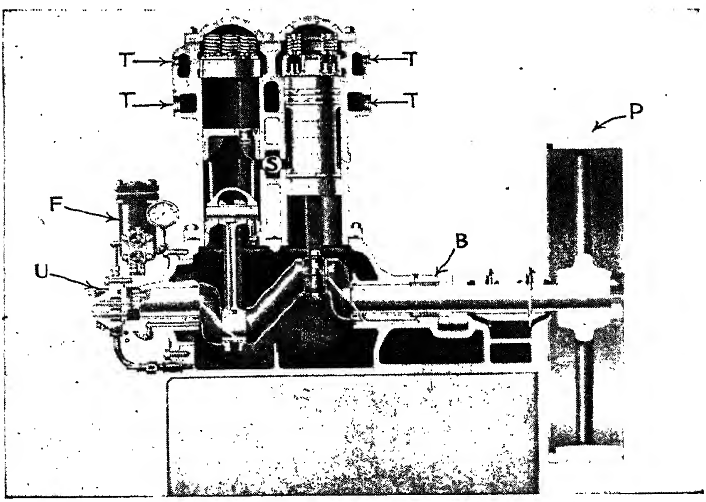


FIG. 76.—Twin single-acting compressor.

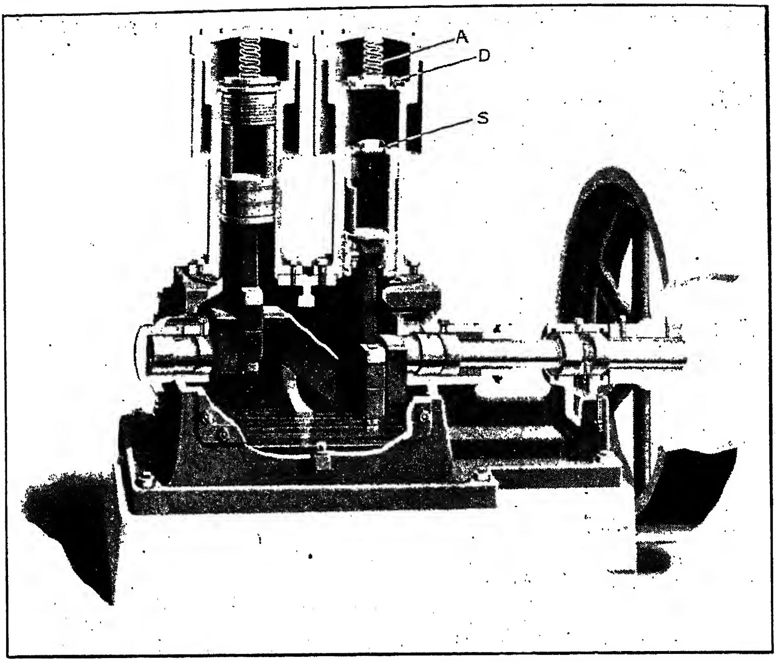


FIG. 77.—Cross-section of vertical compressor with single ring-plate valves (Figs. 67 and 68).

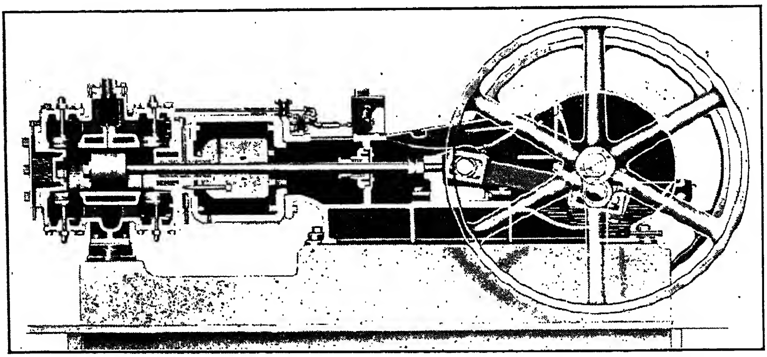


FIG. 78.—Cross-section of ammonia compressor with stuffing boxes, as shown in FIG. 62, making an enclosed crank-case for lubrication.

Design details of a twin single-acting compressor are shown in Fig. 76. The suction intake is marked *S* and the discharge valves are shown at the top of each cylinder. The stuffing box with

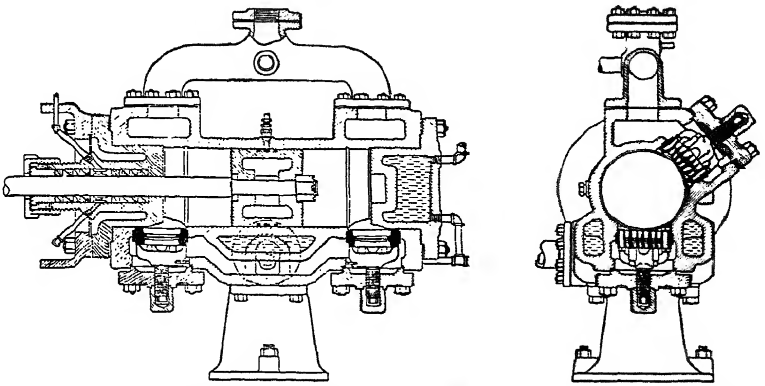


FIG. 79.—Cross-sections of high-speed horizontal ammonia compressor with strip-plate valves.

its connections for oiling is at *B* and the connections for the water jackets at *T*. The driving pulley is at *P*, and the oil pump *U* is located on the end of the shaft.

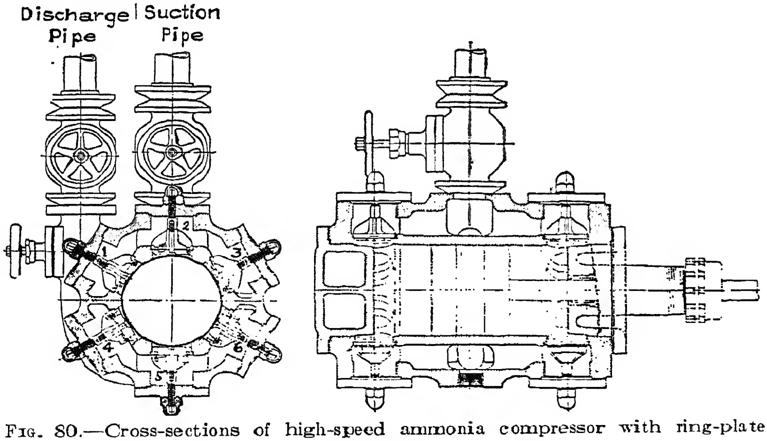


FIG. 80.—Cross-sections of high-speed ammonia compressor with ring-plate valves.

Double-acting Ammonia Compressors.—A double-acting compressor generally has its cylinder in a horizontal position. In the double-acting compressors shown in Figs. 79 and 80, the discharge

and suction valves are attached to the cylinder. The head of the cylinder shown in Fig. 79 is shaped to fit the nut on the piston. The cylinder head is made of this shape so that the clearance space may be small.

Medium-speed Horizontal Ammonia Compressors.—Figure 58 shows, in some detail, the construction of a medium-speed horizontal double-acting ammonia compressor. This compressor has spherical-shaped heads and a piston with faces of the same shape.

The suction manifold, valves, and water jacket are marked in the figure. The valves are of the poppet type and are at the ends of the cylinder, the suction valves being above the discharge valves.

High-speed Horizontal Ammonia Compressors.—High-speed ammonia compressors have been developed for the purpose of making it possible to operate them by direct connection to electric motors and high-speed oil and gas engines. Such direct connection is very desirable, because, in the first place, it simplifies operation and makes unnecessary the use of belt and chain drives with a considerable saving of space in the plant. Provision is made in cylinder heads for water cooling, as these are the places where cooling has the most effect.

Figure 80 shows a horizontal compressor of the double-acting, return-flow type. There are three suction valves marked 1, 2, and 3 on the top side of the cylinder at each end, and, similarly, there are three discharge valves marked 4, 5, and 6 on the bottom side at each end. The valves are of the *ring-plate* design.

A compressor with valve arrangements very different from any of those described above is shown in Fig. 79. This is a horizontal compressor of the double-acting return-flow type, which has valves of the *strip-plate* design, sometimes called *feather valves*. Suction and discharge pipes are indicated for each compressor. The three types of ammonia compressors shown by Figs. 73, 79, and 80 are representative of the installations in a majority of the most recently equipped refrigerating plants.

An oscillating-piston type of compressor is shown in Fig. 36, with description on page 55. A centrifugal rotary type of compressor with five stages of compression is shown in Fig. 85 (p. 132).

Dual or Multiple-effect Compressors.—In some types of refrigerating plants the requirements for refrigeration are such as to require two different operating temperatures and, therefore, different operating pressures. In such cases, it is possible

to "split" a vertical twin compressor using one side for the higher and the other side for the lower suction pressure. This same operating principle can be accomplished in the *dual* or *multiple-effect* compressor, which is shown in Fig. 81. A compressor for this service should be built so as to displace the proper amount of vapor at the lower pressure and should be so designed that at the end of the suction stroke a valve is opened to the evaporator at the higher pressure, after the valve controlling the flow of low-pressure vapor is closed, permitting the pressure in the cylinder to rise to the higher suction pressure. The higher suction-pressure valve is, of course, closed at the beginning of the compression stroke, thus entrapping some of the lower-pressure vapor as well as the higher-pressure vapor. In this type of compressor the piston displacement is smaller than would be needed for the ordinary type, and the horsepower expended is somewhat reduced. The saving in power is also effected by the difference between the two operating pressures. It is essential that a large portion of the operating load should be carried at the higher pressure. This type of compressor is adapted to ice plants which precool both the water for the ice tank and the liquid refrigerant supplied to the freezing tank.

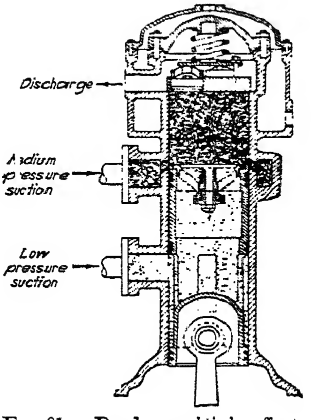


FIG. 81.—Dual or multiple-effect compressor.

Carbon-dioxide Compressors.—Refrigerating compressors for operation with carbon dioxide differ little from those built for ammonia, except in the use of small cylinders and the necessary special provisions for very high pressures. Such compressors are made with vertical or horizontal cylinders and are generally double-acting. The chemical inertness of carbon dioxide permits the use in construction of some metals, as, for example, copper and its alloys, which cannot be used in contact with ammonia when water vapor may possibly be present.

In Fig. 82, the cylinder of a vertical, double-acting carbon-dioxide compressor is shown. This cylinder is made of a steel forging and is bored out of a solid block of metal. The small diameter of the cylinder makes it possible to produce very high

pressures without causing excessive forces in the piston rod, connecting rod, and crank.

The carbon-dioxide gas enters the cylinder C through the passage A and the suction valves B, B. The gas is compressed in the cylinder by the piston P and is expelled after compression at a very high pressure through the discharge valves D, D into the discharge passage E.

The piston is fitted with metallic piston rings, while cup leathers are used as packing in the stuffing box F.

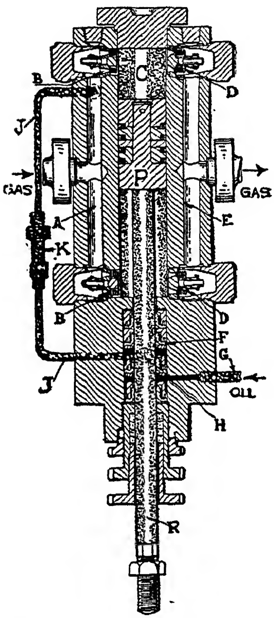


FIG. 82.—High-pressure carbon-dioxide compressor.

Lubrication of this type of compressor is a special problem. A mechanical force-feed lubricator delivers oil through the pipe connection G to the lantern[1] of the stuffing box. From the point H, in the inner section of the stuffing box, a pipe J leads to the suction passage A through a check valve K. This pipe permits any carbon-dioxide gas leaking into the stuffing box along the piston rod R to return to the suction valves of the compressor. With this gas is carried any oil which has found its way from the oil lantern H. The distribution of this oil on the cylinder walls provides the necessary lubrication and sealing-oil film over the piston rings. In a vertical-cylinder compressor, the pipe J extends upward from the stuffing box and enters the suction chamber or port above the center in order that the oil may pass into the cylinder through the top valve from which it is distributed downward by the piston. In horizontal-cylinder compressors, this connection enters the suction passage or port centrally.

Volumetric Efficiency and Clearance of Compressors.—The ratio of the volume of the vapor of the refrigerant which is actually handled to the volume of the piston displacement[2] is called

[1] The *lantern* is a space in the central portion of a stuffing box, not filled with packing into which the gas leaking through the packing can accumulate.

[2] *Piston displacement* is the volume of the space in a cylinder swept through by the piston. It is calculated by multiplying the area of the piston ($3.1416 \times D^2 \div 4$, where D is the diameter of the piston) by the distance the piston

the *volumetric efficiency*. It is customary when calculating volumetric efficiency, to express the volume of the vapor of the refrigerant which is actually handled in terms of the conditions of temperature and pressure in the cooling coils of the evaporator. For example, if the actual volume of the ammonia vapor taken into the cylinder of a compressor in 1 minute is (0.422 pound per ton of refrigeration) 3.44 cubic feet per ton and the piston displacement is 0.1 cubic foot per stroke for a 5-ton double-acting compressor, making 200 strokes per minute (100 revolutions per minute), the piston displacement per minute is 200×0.1 or 20 cubic feet per minute, and the volumetric efficiency is $3.44 \times 5 \div 20$, which is 0.86 or 86 per cent.

There are a number of reasons why the volumetric efficiency of a compressor is less than 100 per cent: (1) The amount of vapor of a refrigerant that will fill the volume of the piston displacement at the temperature and pressure in the cooling coils of the evaporator is always greater than the amount actually taken in. (2) The refrigerant vapor is superheated by throttling when it enters the cylinder of the compressor, and this heating by throttling produces a higher temperature than that corresponding to the pressure; in other words, if the vapor is dry and saturated in the suction pipe, it will be superheated in the cylinder of the compressor. This superheated vapor of the refrigerant, being warmer than the vapor in the cooling coils of the evaporator, is obviously less dense and weighs less per cubic foot. (3) Another condition tending to reduce the actual volume of the vapor of the refrigerant that enters the cylinder is the lowering of density of the vapor of the refrigerant in the cylinder because its pressure is less than the pressure in the cooling coils of the evaporator. This lower pressure in the compressor cylinder is necessary in order to establish a flow of vapor from the cooling coils of the evaporator into the cylinder. This flow of vapor must occur necessarily at high velocity because of the rapidity with which the cylinder must be filled at each stroke of the compressor. (4) Still another reason for the reduced volume of the refrigerant entering the compressor cylinder is that there is a larger space inside the cylinder than is actually swept through by the piston. This "excess" volume is called the *clearance*. During the discharge stroke of a compressor, all of the compressed vapor of

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moves in a stroke of the compressor. When the dimensions for calculating displacement are in inches, the cylinder displacement is in cubic inches.

the refrigerant is forced out of the cylinder through the discharge valve as the piston in its cylinder advances on the stroke, with the exception of the vapor which is trapped in the *clearance space.* This trapped vapor remains and, on the next suction stroke of the compressor, expands down to the pressure in the cooling coils of the evaporator and fills a portion of the cylinder displacement, thus *limiting the amount of cylinder volume* that is available for the new charge of refrigerant. The higher the discharge pressure the more objectionable the clearance becomes.

Clearance of Compressor.—An important detail in the construction of a compressor is its clearance. The clearance of a compressor is the space between the piston and the cylinder head when the piston is at the end of its stroke. A small amount of clearance[1] is necessary, as the piston must be prevented from coming up against the cylinder head and doing damage. On the other hand, if the cylinder has a clearance too large, there will be an excessive amount of compressed vapor left in the cylinder at the end of the compression and discharge stroke, and then, on the next suction stroke, the vapor which is left in the cylinder at high pressure expands until its pressure falls to the suction pressure. At this point, the suction valve opens, and the cylinder begins to take in a new charge. The expansion of the vapor left in the cylinder reduces the available space for the new charge, a condition that greatly reduces the capacity of the compressor. The clearance space in a compressor should, therefore, be as small as it may be with safety, not exceeding 5 per cent of the cylinder volume.

Volume Delivered by Compressors or Pumps.—In order to determine the volume delivered by a compressor or pump, the piston displacement,[2] which is the volume swept through by the plunger or piston in each stroke, must be calculated.

[1] In some makes of ammonia compressors having vertical, single-acting cylinders, the amount of clearance is made as small as possible without the risk of the piston's coming into direct contact with the head of the cylinder. In some compressors of this kind, the distance between the piston in its highest position and the valves of the cylinder head is not more than $\frac{1}{64}$ inch.

[2] Piston displacement is explained more in detail on p. 126. In some pumps, the total area of the plunger or piston is not effective, because of the attachment of a piston rod. When a piston rod is used, reducing the effective area of a plunger or piston, the area of the rod must be deducted from the total area of the face of the plunger or piston.

Volume Delivered by Single-acting Compressor.—If a compressor makes 50 strokes per minute, then the plunger or piston displacement per minute is found by multiplying the plunger or piston displacement per stroke by 50. A standard United States gallon is 231 cubic inches, so that the *displacement per minute* expressed *in gallons* is the plunger or piston displacement per minute in cubic inches divided by 231. There are always some leakage and other losses in a compressor, so that the actual volume delivered will be somewhat less than the theoretical displacement as explained above. The losses are due to incomplete filling of the cylinder of the compressor at each stroke and leakage through the valves and around the plunger or piston.

Volume Delivered by Double-acting Compressor.—The displacement of a double-acting piston type of compressor is calculated in very much the same way as the displacement of a single-acting compressor or pump. The method of calculating the displacement of a double-acting piston type may be explained by taking the case of a compressor with a cylinder 15 inches in diameter, a piston stroke of 30 inches, a piston rod 3 inches in diameter, the compressor making 120 strokes per minute. The area of the piston on the side where there is no piston-rod area to consider is 176.7 square inches. The area of the piston rod is 7.1 square inches. Deducting 7.1 from 176.7 square inches, the effective area of the side of the piston to which the piston rod is attached is 169.6 square inches. Since there are 120 strokes of the piston per minute, half this number, or 60, are effective for the discharge of compressed gas or vapor at one end of the cylinder, and the other 60 strokes for the discharge at the other end. The displacement of the compressor *per minute* at the end with the larger piston area is $(176.7 \times 30) \times 60$ or 318,060 cubic inches, and the displacement of the compressor *per minute* at the other end is $(169.6 \times 30) \times 60$ or 305,280 cubic inches. The total displacement per minute is 623,340 cubic inches per minute.

Gear Compressors.—About 10 years ago a gear compressor was built by the Isko Company. This was small being constructed in small sizes for household units. More recently the Sturtevant Company has developed a gear compressor which consists of special gears of the herringbone type. This type of gear is used to avoid pulsations while the refrigerant vapor is being gradually delivered as the teeth reach the high side of the compressor casing.

The gear compressor consists of two herringbone gears as shown in Fig. 83, one being driven by the shaft of which it is an integral part. These gears are each cast in one piece, and the teeth are of the ordinary involute type. A double-ring ball bearing is used at each end of the gears. The small clearance space between the ends of the teeth and the end plates is filled with oil when in operation, in order to hold the refrigerant vapor in the high side of the casing. The vapor passes from the bottom to the top. The bearings and gears are constantly supplied with lubricant under pressure. The lubricant is cooled before returning to the casing.

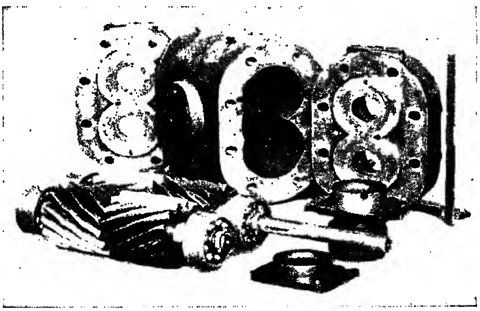


FIG. 83.—Gear compressor.

Laboratory tests of large-gear compressors have shown higher volumetric efficiencies than are obtained with reciprocating compressors. In some cases a power consumption of from 0.6 to 1.2 horsepower per ton of refrigeration is attained. The rotative speed is 1,750 and 1,150 revolutions per minute for the 15- and 60-ton units, respectively. These units are built in dual with an electric motor located between the compressors and directly connected, making it possible to construct large compressors by multiple steps of 60 tons. The lubricant is carried into the condenser where it is separated from the refrigerant.

Rotary Compressors.—This type of compressor has long been in use as a device for pumping against pressures as high as 80 to 90 pounds per square inch. In some designs it consists of two rotating elements variously called "pistons," "impellers," or "lobes." The rotation of the shafts and impellers traps the vapor of the refrigerant between the lobes and the case and delivers it to the condenser at the required pressure. The rolling

together of the impellers, as in the case of the gear compressor, prevents the return of the refrigerant vapor to the low side of the system. A compressor having one eccentric rotating element located in the casing and provided with a number of packing blades which are free to slide radially in the rotor is shown in Fig. 84.

Centrifugal Compressor.—A centrifugal compressor is a distinctly different type of machine from the gear or rotary type of compressor, these being positive types of compression machines. The centrifugal compressor produces a pressure solely by its centrifugal effect; that is, by the kinetic energy of the particles of a revolving mass of refrigerant. In all other types of compressors, the particles are actually squeezed together within the casing by the positive action of the pistons, gears, or lobes; the compression being produced intermittently by the operation of the suction and discharge valves. In the centrifugal compressor, there is no wear on valves and moving parts other than the outside bearings supporting the shaft. It is not subject to corrosion or erosion, and good efficiencies may be obtained permanently. Commercial centrifugal compressors have been designed to give mechanical efficiencies ranging from 70 to 80 per cent.

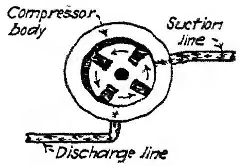


FIG. 84.—Rotary compressor with packing blades.

The fundamental design of the centrifugal compressor is identical with that of a centrifugal pump. However, the centrifugal compressor uses much higher peripheral speeds than centrifugal pumps. Metal labyrinths[1] are used to prevent the leakage of refrigerant between the various stages; this leakage, if permitted, would lower the efficiency. Another important feature of centrifugal compressors is that no internal oiling is necessary as there are no internal friction surfaces Because of this, the heat-transfer surfaces are never fouled by a heavy accumulation of oil as is the case with other types of compressors. In addition to the above advantages there are two others: (1) operation at a fixed speed to "handle" a variable amount of refrigerant vapor, and (2) small space for a given capacity.

Figure 85 is a view of a typical centrifugal compressor with the top-half of the casing removed to show the inlet and discharge

[1] Labyrinth packings are explained in detail in "Steam Turbines" by James A. Moyer, John Wiley & Sons, Inc., New York.

passages (marked, respectively, 1 and 6) and the five impellers or rotors, each of which is in a separate compartment or so-called stage, the compressor in this case being, therefore, a five-stage design. The vapor of the refrigerant, which is to be compressed, enters the compressor through the suction passages marked (1, 1) and passes into the largest of the impellers through openings near the shaft. By the centrifugal force developed in the impeller, the vapor is discharged from its periphery at a higher pressure than it had when entering. The vapor thus compressed in

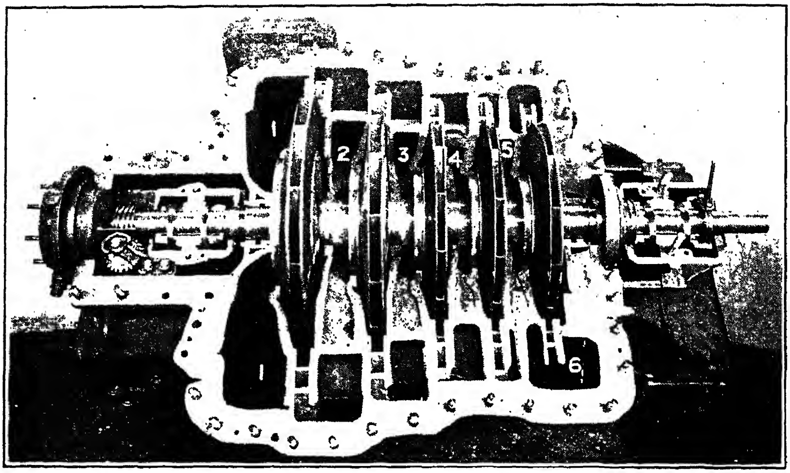


Fig. 85.—Five-stage centrifugal compressor.

the first impeller discharges into the space marked 2 between the first and second impellers, passing through openings near the shaft of the second impeller to be discharged from it into the space marked 3. In this way the vapor traverses a somewhat irregular path through the centrifugal compressor to be discharged finally into the high-pressure discharge space marked 6. A centrifugal compressor is designated by its number of stages, there being one stage for each impeller. The compressor shown in the figure has; therefore, five stages.

The Brown-Boveri Company of Switzerland has built a centrifugal compressor having a capacity of 2,700 tons of refrigeration which requires less space than one of the conventional reciprocating-piston type of 1,000 tons. Another machine of this type has been built which develops 20 tons of refrigeration at a suction

temperature of 22° F. below zero and operates at a speed of 18,000 revolutions per minute. Centrifugal compressors with six stages operating with sulphur dioxide as the refrigerant have been designed to deliver 500 tons of refrigeration at a speed of 5,000 revolutions per minute, the impeller having a diameter of 22.92 inches.

The following table shows the minimum capacity of centrifugal refrigerating units for various refrigerants, based on the speed with which the impellers can be safely rotated and conveniently manufactured.

	Capacity of Refrigeration, Tons
Carbon dioxide	1,000
Ammonia	480
Methyl chloride	200
Sulphur dioxide	125
Butane	100
Ethyl chloride	83
Carrene	25

According to one authority[1] if one stage is required for carbon dioxide, the *relative* number of stages needed for other refrigerants is as follows: ethyl chloride, 1.23; sulphur dioxide, 1.26; methyl chloride, 1.3; ammonia, 4.3.

Adiabatic Compression.—When the vapor of a refrigerant is compressed or expanded without a loss or gain of heat (from another source), the compression or expansion is called *adiabatic*. In the analysis of what takes place in the cylinder of a compressor in a refrigerating system, it is assumed that the compression is adiabatic, meaning, there is no transfer of heat between the vapor of the refrigerant and the cylinder walls, just as if the compression took place in a cylinder made of a material which is a non-conductor of heat. In many of the problems which are to be calculated in this subject, it will be assumed that the vapor of the refrigerant enters the cylinder of the compressor during the suction stroke as a dry and saturated vapor, that is, in the form of a vapor containing no particles of any of the *liquid* refrigerant which may be in the cooling coils of the evaporator; in other words, the vapor of the refrigerant is not superheated at the *beginning* of the compression stroke. In most cases, however,

[1] *Refrigerating Engineering*, Vol. 20, No. 6, p. 37.

at the *end* of the compression stroke, the vapor of the refrigerant is superheated to a considerable degree, meaning that the temperature is a number of degrees above the saturation temperature corresponding to the pressure.

Pressure after Compression.—In a mechanical refrigerating system, the pressure to which the vapor must be compressed depends on the temperature of the cooling water supplied to the condenser. In a system using ammonia as the refrigerant, the temperature of the cooling water should be between 50 and 80° F. In the ammonia table on page 494, it will be seen that, under these temperature conditions, the ammonia vapor must be compressed to some value of absolute pressures between 89.16 and 153.0 pounds per square inch or, in gage pressures, between 74.49 and 138.3 pounds per square inch.

Pressure before Compression.—When liquid ammonia evaporates at atmospheric pressure, the temperature of the ammonia is 28° *below* the Fahrenheit zero (−28° F.), a temperature much below that required for commercial refrigeration. The suction pressure of the compressor in most commercial ammonia refrigerating plants is, therefore, greater than atmospheric. The *suction pressures* generally used in practice are from 20 to 50 pounds per square inch absolute. With this pressure in the low-pressure piping, air leakage is prevented, and the size of the compressor can be reduced for given requirements.

CHAPTER V

HOUSEHOLD MECHANICAL REFRIGERATION

Modern Household Refrigeration Systems.—Automatic mechanical household refrigeration was introduced about 1910. Before this time, the ordinary ice-cooled refrigerator was the common device for preserving and storing foods in the home. The development and use of the electrical refrigerator have been rapid. According to statistics of the U. S. Census, there are between 20,000,000 and 30,000,000 families in the United States, of whom about one-third live under more or less urban conditions, which are distinctly favorable to extensive use of mechanical refrigerators. Since 1920, the equipment of homes with these devices has progressed steadily. Fortunately for the industry, household machines have been brought to such a stage of perfection that the amount of servicing has constantly diminished.

A study of various foods shows that at temperatures ranging from about 40° to 50° F. the bacteria multiply about one-four-hundredth as rapidly as in the range between 50 and 60° F. From this it can be seen that a household refrigerator should cool the stored foods to temperatures below 50° F. This temperature is rather difficult to obtain in an ice-cooled refrigerator.

Fungi may be classified as follows: (1) molds, (2) yeasts, and (3) bacteria. The growth of bacteria can be shown by the fact that one bacterium will produce, after 1 hour, 4 bacteria; after 2 hours, 16 bacteria; after 5 hours, 65,543 bacteria; and, after 15 hours, 1,000,000,000 bacteria.

Household refrigerating systems may be divided into the following two classes: (1) the *compression system*, which is electrically driven, such as the Frigidaire, Kelvinator, Servel, and similar makes; and (2) the *absorption system*, in which the necessary energy is supplied from either an electric heater or a gas burner, as in the commercial absorption system. There is further explanation of this system on pages 190–202.

The advantages of the compression system are that the pressures involved need not be very high and either air or water

may be used for condensing the refrigerating medium. The disadvantage is that a somewhat noisy belt or gear drive is generally used, while the electric or gas-heated absorption system is practically noiseless, as it has no moving parts. The principal disadvantage of the absorption system is that high pressures are necessary as most systems use ammonia as the refrigerant, and a supply of water is generally required for cooling purposes, a fact which considerably increases the operating expense of small machines.

The following table shows the comparative operating cost for these systems:

Compression System (Electric)		Absorption System (Gas Heated)	
Average electric-power consumption 50 kilowatt-hours per month		Gas consumption....	1,500 cubic feet per month
Cost of electrical power:		Cost of gas..........	$1.15 per 1,000 cubic feet
First 20 kilowatt-hours.............	8.5 cents		
All over 20 kilowatt-hours.............	3.0 cents	Water consumption..	5 to 8 gallons per hour or 500 to 900 cubic feet per month
Total cost of operation.	$2.60 per month		
		Cost of water........	$0.75 to $2 per 1,000 cubic feet
		Cost of gas..........	$1.75 per month
		Cost of water........	$0.35 to $1.80 per month
		Total cost of operation.	$1.10 to $3.55 per month

From the above table it can be seen that the average operating cost is approximately the same in both systems. These figures apply to cabinet refrigerators having a capacity 6½ cubic feet and a "cooling" temperature of 40 to 45° F. inside the box. The ice-melting capacity is, in each case, about 75 pounds of ice in 24 hours, so that the *equivalent* cost of refrigeration in terms of "melted" ice would be about 12 cents per 100 pounds without considering depreciation and interest.

Ice Refrigerators.—The most common refrigerant used for household purposes is ice. It has been estimated that the yearly consumption per capita is about 1,000 pounds. As the melting

point of ice is 32° F. the cabinet temperature will be about 42° F. at the bottom and 52° F. at the top for a well-filled ice compartment and a well-insulated cabinet, these temperatures varying with the outside temperature of the air. The heat transfer is chiefly by convection (p. 373), the circulating air being the carrier of heat. Since only a small percentage of the circulating air comes in contact with the ice, the surface area of the ice should be at all times kept large to maintain low box temperatures. The refrigerator cabinet should be carefully designed to provide adequate and free circulation of the air in the food compartment

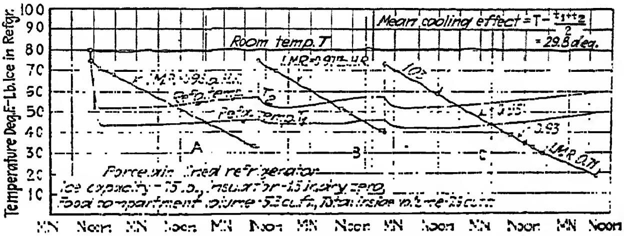


FIG. 86.—Test results of moderate-priced 75-pound ice refrigerator.

as well as through the ice chamber. The cabinet should be insulated with about 2 to 3 inches of cork board or other equivalent insulation. The baffles and partitions should be well insulated to insure good air circulation. A suitable water seal to prevent the outward flow of cold air from the cabinet should be provided, while the doors should be equipped with proper gasket material.

The ice-melting rate, cooling effect, and temperature differential are all criteria of performance. The test results for a moderate-priced 75-pound refrigerator are shown in Fig. 86.[1] The test was made in a room held at 80° F., and with a 60 per cent charge of ice. The figure shows the high and low (T_2 and T_1) temperatures taken at the top and bottom of the cabinet for the period of the test, the cooling effect being 29.8° F., which is the difference between the outside air temperature and the average inside cabinet temperature. The temperature difference (p. 280) was 10° F. The ice-melting rate per hour was 0.96 pound.

In the case of an air-insulated box, typical of more than 80 per cent of the refrigerators sold today, the test showed the cool-

[1] FRAZIER, R. T., "Factory Testing of Refrigerators," *Refrigerating Eng.*, Vol. 20, No. 3, pp. 159–163, September, 1930.

ing effect to be 23.2° F. and the temperature differential 10° F., while the ice-melting rate was 0.813 pound per hour.

The use of ice baskets has been carefully studied.[1] These baskets permit a freer circulation of air, thus improving the box performance for a given ice consumption. The baskets are made of perforated metal or wire mesh. Broken or chunk ice may be used to fill the basket. The ice-melting rate was found from tests to be less than when the refrigerator was cooled without the baskets, and the cooling effect with the baskets was more uniform than without them.

The humidity of the outside air affects the humidity in the cabinet; that is, as the outside humidity rises, the box-humidity in the cabinet likewise rises—but not so rapidly. This change in humidity affects the ice-melting effect, there being a greater ice consumption for increased humidity.

Refrigerants for Household Systems.—The refrigerants used in household refrigerating systems are ice, sulphur dioxide, ethyl chloride, methyl chloride, ammonia, carbon dioxide (not commonly used in America but used extensively in Europe), butane, isobutane, and dichlorodifluoromethane. These refrigerants can be classified into two groups: (1) non-inflammable and (2) inflammable. The non-inflammable refrigerants are carbon dioxide, sulphur dioxide, and dichlorodifluoromethane. The remainder of the group may burn when mixed in some proportions with air and must, therefore, be classified as inflammable. Not all of the above refrigerants are widely used in household machines. Those most commonly found are sulphur dioxide, methyl chloride, dichlorodifluoromethane, and ammonia. Isobutane and butane are used to some extent.

General Electric Company Refrigerating Unit.—The General Electric household refrigerator has been designed to occupy as little space as possible and to eliminate all exposed moving parts. It has been arranged to simplify the interchange of refrigerating units and to reduce to a minimum the possibility of gas leaks. An automatic control maintains constant refrigerating temperature. The refrigerant used in this machine is *sulphur dioxide.*

The General Electric unit resembles in many ways the Audiffren oscillating-cylinder refrigerating machine (p. 55), which has been successfully used for 25 years. There are five principal

[1] BELSHAW, C. F., "Ice Baskets for Domestic Refrigerators," *Refrigerating Eng.*, Vol. 20, No. 5, pp. 291–294, November, 1930.

parts in the refrigerating unit: (1) compressor; (2) float valve; (3) evaporator; (4) automatic temperature control; (5) condenser. The parts are marked in Fig. 87.

Compressor.—The compressor of the General Electric unit is shown in Fig. 87. It has a single oscillating cylinder, and its piston is driven by an eccentric on the shaft of the electric motor.

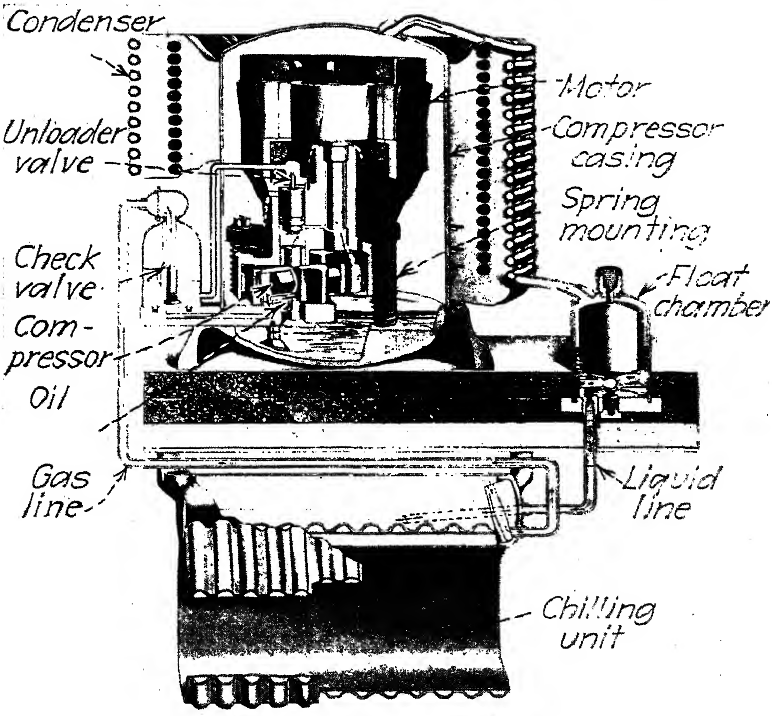


FIG. 87.—Diagram of General Electric icing unit.

The compressor is single-acting, being made with a 1-inch piston and a stroke of 0.7 inch. The compressor and motor are in a steel case which is provided with a steel base plate. The joint between the case and the base plate is made thoroughly leak-proof by means of a tongue-and-groove type of lead seal. Lubrication is by the forced-feed method that operates by means of a plunger type of oil pump which operates on the permanently sealed oil supply, somewhat as the piston of the compressor operates on the refrigerant. The oil pump is shown in the figure.

In order to reduce the starting torque of the compressor, an "unloader" valve is used. This valve is held up against its seat by oil pressure during the normal operation of the compressor but opens at low oil pressure when stopping, thereby allowing the pressure on the outside and the inside of the compressor cylinder to become equalized through a by-pass. At the time of stopping, a check valve closes and thus prevents the vapor at high pressure from leaking back into the evaporator through the suction line.

The entire mechanism is mounted within the steel case on three springs, so as to absorb motor noises and vibrations. This makes the refrigerating unit practically noiseless.

In order to reduce the viscosity of the oil and reduce the motor-starting torque, a little device called an "oil conditioner" is connected directly across the main electric line. This oil conditioner or heating element consists of a fine nichrome-steel wire embedded in porcelain and is made so that it is easily replaced through a hole located in the drop-forge steel base. The heater element draws about 10 or 12 watts at 110 volts. This input is not, however, an additional load as the over-all actual input to the unit is increased only 5 watts. The oil conditioner also serves to drive off any sulphur dioxide held by the oil thus eliminating the sulphur-dioxide condensate from accumulating in the base.

Float Chamber.—The float chamber is located on the top of the refrigerator cabinet to one side of the compressor case and condenser. Its purpose is to separate the high- from the low-pressure sides of the system, and to accumulate the liquid refrigerant. When there is a sufficient quantity of liquid in the float chamber, the float valve[1] lifts and allows the liquid refrigerant to flow into the chilling unit or evaporator.

Evaporator or Chilling Unit.—The evaporating device is located on the inside of the cover of the cabinet and is an integral part of the unit. It is made of two steel sheets, one of which is corrugated. These sheets are folded into shape with the upper part of the inner and outer sheets forming a cylinder-shaped header. The sheets are brazed and welded together electrically. This construction produces in effect a series of parallel tubes extending around the outer surface of the chilling unit. These tubes open into the refrigerant reservoir. In the central recess of the evaporator are two or more trays. These trays may be used to make ice cubes or frozen desserts.

[1] The adjustment of the float valve is made at the factory.

Control of Temperature.—The temperature-regulating device is located in a central box placed on the top of the refrigerator cabinet to the left of the compressor, as shown in Fig. 88. This control performs three functions: (1) Starts and stops the electric motor as the temperature changes in the evaporator; (2) cuts off the current whenever there is an overload, thus preventing damage to the motor; (3) varies the temperature of the cabinet as desired.

The temperature-control mechanism is shown in Fig. 89 and consists of a "sylphon bellows" to which a copper tube is attached,

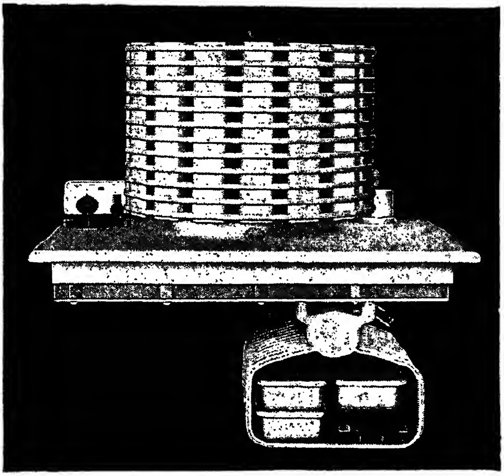


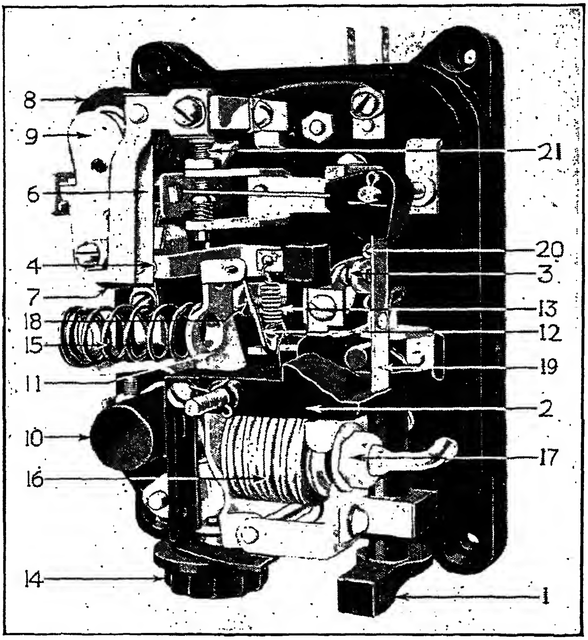
Fig. 88.—General Electric icing unit showing parts in place.

the other end of which is in contact with the evaporator. This copper tube is filled with sulphur dioxide and is not in any way connected to the sulphur dioxide in the main unit. As the temperature in the cabinet rises, the pressure of the sulphur dioxide in the tube rises and likewise the pressure in the sylphon bellows, forcing the bellows to expand and close the switch which starts the motor. On the other hand, a decrease in temperature in the evaporator causes a reduction in pressure in the sylphon bellows which causes the switch to open, cutting off the electric current and stopping the motor.

A dial with a pointer is provided so as to decrease or increase the tension of the temperature-control mechanism to be set for a

range of 20° F. In order to prevent damage from overload, an overload heater element trips the overload cutout element which opens the electric circuit.

Electric Circuit.—The electric circuit for the General Electric icing unit is shown in Fig. 90. It should be noted that in the



1. Main switch
2. Latch and indicating arm
3. Main contacts
4. Starting contacts
6. Starting contact spring
7. Starting relay series coil
8. Starting relay shunt coil
9. Starting relay armature
10. Starting resistor
11. Lever for automatic control
12. Bridle
13. Bridle spring for contact arm
14. Temperature adjusting knob
15. Temperature adjustment spring
16. Metallic bellows
17. Clamp nut on bellows
18. Temperature-range adjusting scr
19. Overload cutout
20. Overload heater
21. Overload adjusting screw

FIG. 89.—Temperature-control mechanism.

design of this unit there must not be any electrical contacts within the hermetically sealed casing. The induction motor if it is to run on a single-phase 60-cycle 110-volt circuit must have a special wiring circuit other than that commonly used with split-phase induction motors for starting.

This induction motor is provided with two windings, namely, the starting and running windings. The starting winding is to

be cut out of the circuit after the motor is up to speed. This necessitates the use of a series starting coil and a shunt starting coil, the starting shunt coil being cut out of the circuit by the series coil opening the starting contacts when the motor has come up to its normal speed. The main contacts are closed and opened

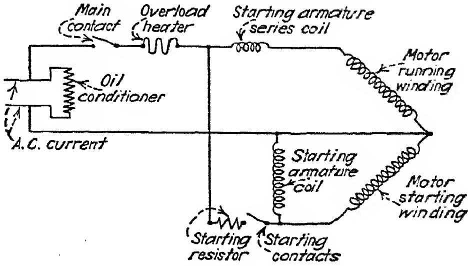


FIG. 90.—Wiring diagram of General Electric refrigerator.

by the pressure of the sulphur dioxide in the sylphon bellows and copper tube.

The starting operation is as follows: The sylphon bellows closes the main contacts, and the electric current flows into the running winding and starting winding as the starting contacts are closed. The starting resistance being located in the starting circuit causes a sufficient phase displacement between the two currents in the two windings to start the motor against its starting torque. As the current reaches a maximum in the running winding, the series coil opens the starting contacts, cutting out the starting winding. When the main contacts open the armature of the starting coil returns to its starting position, closing the starting winding-circuit contacts, so that the circuit is in its normal starting condition.

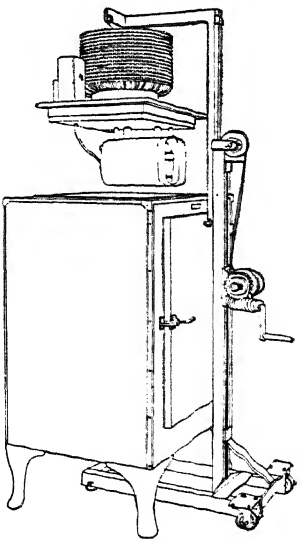


FIG. 91.—Portable crane for removing General Electric icing unit.

In some districts of our cities direct current is used and because of the fact that the General Electric icing unit has an induction motor it is then necessary to convert the direct current into alternating current.

This is accomplished by the use of a small rotary converter which is designed to operate on a 110-volt direct-current line.

The installation of this refrigerating unit consists only of placing it in position at the top of the cabinet; and since there are no pipe connections to be made, it is easily installed in an apartment or a house. Figure 91 shows a crane for conveniently

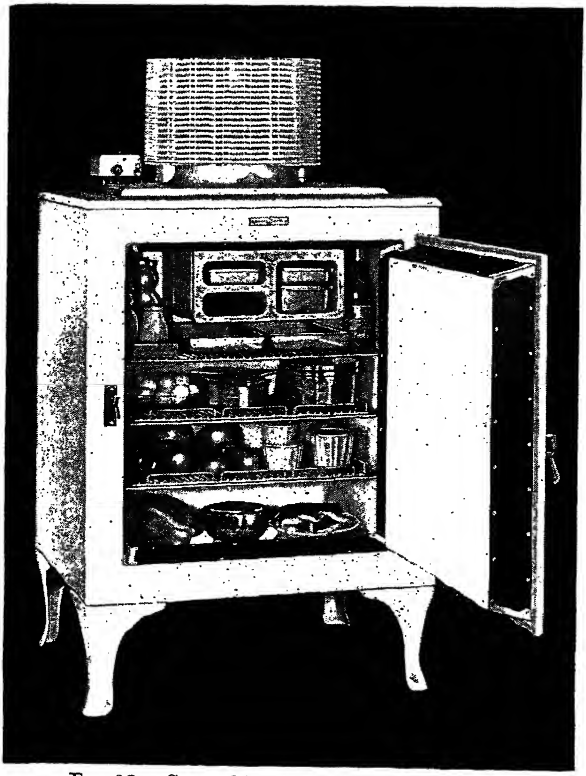


FIG. 92.—General Electric refrigerating unit.

removing and replacing this unit in a refrigerator. A typical household refrigerating unit is shown in Fig. 92.

At the factory, the standard method of testing this machine is to submerge the evaporator in a brine bath held at a temperature of 20° F. while the condenser for the refrigerant is at room temperature. With these conditions, one of the small units has a refrigerating capacity of 320 B.t.u. per hour, when the electric-

power input is 150 watts and the room temperature is 100° F. The condenser gage pressure is 110 pounds per square inch while the suction pressure corresponds to a vacuum of 4 inches of mercury.

When this refrigerating unit is installed in a room having a temperature of 100° F., it will run about 70 per cent of the time when the doors of the refrigerator are kept closed. Under this condition, the average suction pressure is slightly lower than during the brine test method above, and the refrigerating capacity of the machine will be slightly reduced.

The following table gives the capacities of the General Electric refrigerating units operated in a room held at 80° F. and at a chilling unit temperature of 20° F.

Model	Capacity, B.t.u. per hour	Size of motor, horsepower	Speed, revolutions per minute
DR-1	300	1/10	1,740
D-2	420	1/8	1,740
D-35	670	1/6	1,740
DR-4	1,400	1/3	1,740
D-50	2,100	1/2	1,740

Frigidaire Compression Refrigerating System.—The Frigidaire system made by the Frigidaire Corporation (General Motors Corporation) is of the compression type and operates according to the following cycle: The heat is absorbed from the refrigerating cabinet by the evaporating refrigerant, which is *dichlorodifluoromethane*, which hereafter will be known as "F-12," and is carried away by the cooling water or air, whichever cooling medium is used for the condenser. The compressor, driven by an electric motor, serves to keep the refrigerant circulating through the system and increases the pressure of the refrigerant so that it may be readily liquefied in the condenser. The F-12 which has been condensed flows into the liquid receiver from which, by the difference in pressure, it is forced through a tube into the cooling coil of the evaporator. The flow of the liquid F-12 into the cooling coils is controlled by an expansion valve of the float type, as shown in Fig. 93. This valve serves two purposes: (1) to maintain a pressure on the liquid line so as to keep the F-12 in the liquid state at "room" temperatures, and (2) to allow the liquid F-12 to flow into the cooling coil of the evaporator rapidly enough to replenish the refrigerant which has been evaporated

in the coil. The method of operation of this float valve can be seen clearly in Fig. 108. The evaporation of the liquid F-12 in the coil of the evaporator takes place continuously but is more rapid when the compressor is running because the operation of the compressor reduces slightly the pressure in the coil of the evaporator. The float expansion valve (Fig. 93) is located in the float chamber of the cooling unit (Fig. 94) and at all times maintains the proper level of F-12 in the coil of the evaporator,

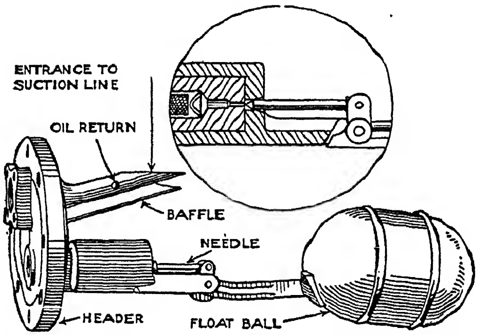


Fig. 93.—Float valve for operating evaporator on flooded system.

regardless of the surrounding temperature or pressure. When the liquid F-12 in the coil evaporates, the float drops, permitting the needle of the expansion valve to open slightly and thus allow additional liquid refrigerant to enter the cooling coil. After the refrigerant enters, the level rises, and the float shuts off the supply. It is easily seen that the control is entirely dependent

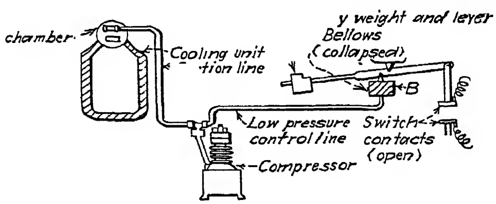


Fig. 94.—Float chamber and low-pressure regulator with sylphon bellows (compressor not running).

on the quantity of liquid in the cooling coil and, in turn, is dependent on the rapidity with which the liquid F-12 evaporates.

Cooling Unit.—The cooling unit, as shown in Fig. 94, consists of a *float chamber* containing the float valve (Fig. 93) and a system of coils of the evaporator designed to bring about the

transfer of heat from the air in the refrigerator to the refrigerant contained in these coils. Each end of these coils is silver-soldered into the wall of the float chamber. The cooling unit does not consist merely of a coil of tubing but has a definite shape for the particular function it has to serve.

Control of Lubrication.—Since the compressor piston must be lubricated to reduce friction, and the lubricating oil must make a perfect seal between the piston and its cylinder to prevent leakage of the vapor, it is a favorable circumstance that F-12 and oil mix well in all proportions. The liquid F-12, when it comes into contact with the oil which has passed through the cylinder of the compressor, mixes with the oil, so that the resulting liquid which passes to the coils of the evaporator is not pure F-12 but a mixture of oil and F-12. The oil used for lubrication

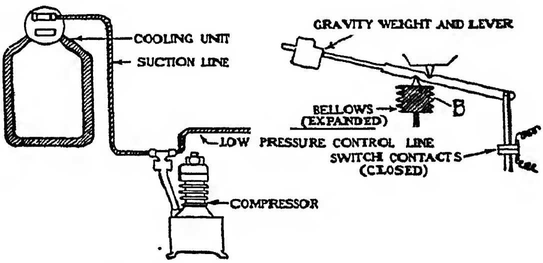


Fig. 95.—Low-pressure regulator with sylphon bellows (in operation).

is returned from the evaporator to the compressor by being carried in a mechanical mixture of the F-12 vapor and the oil, through the suction lines.

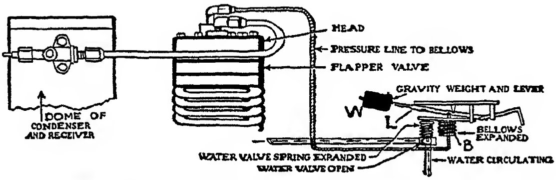
Motor Control.—The unit is equipped with a *low-pressure control,* as shown in Figs. 94 and 95, instead of a thermostatic control, like the device shown on page 165. The refrigerating unit using a low-pressure control maintains the food-compartment temperature by using a surface area of the coils of the cooling unit which is properly proportioned to the refrigerating load.

As the temperature in the food compartment rises, the pressure in the evaporator coils rises also, thus causing the *sylphon bellows B* (Fig. 95) to expand and thus close the electric switch which controls the starting of the compressor. On the other hand, as the temperature in the food compartment drops, there is a lower temperature and pressure in the evaporator coils, the sylphon bellows contracts and opens the electric switch which stops the motor and the compressor. The pressure in the evapo-

rator coils continues to fall until the temperature of the coils is about 8° F.

There are two general types of Frigidaire refrigerating machines. (1) The type commonly used for domestic purposes which has a condenser cooled by air circulation, and (2) the type generally used in commercial installations in which the condenser is cooled by water. The selection of the type of condenser (water or air) is based on the temperature of the available air or water.

Frigidaire Refrigerating System with Water-cooled Condenser.—In the systems where the condenser is cooled with water, some means is needed to control the quantity required. This is accomplished by the sylphon bellows *B* (Fig. 96) which begins to expand when the gage pressure in the condenser is about 112 pounds per square inch. At this pressure, a lever *L* and a gravity



G. 96.—Regulation of cooling water valve by low-pressure control.

weight *W* are raised and cause the water valve to open so that more water is circulated in the condenser. When the cooling water in the condenser is at a comparatively high temperature, the pressure in the condenser will also be high and a large quantity of water will flow. This is necessarily so, because, as the temperature of the cooling water increases, the vapor of the refrigerant has to be subjected to a higher pressure for liquefaction. Expansion of the sylphon bellows produced by any further increase in pressure will cause the water valve to open wider and allow more water to pass through the condenser. It is then easily seen that the supply of water is controlled entirely by the condenser pressure. When the motor and the compressor are stopped by opening the control switch, the pressure in the condenser will gradually become lower. The cooling water, however, will continue to flow through the condenser coils until the pressure in the condenser has been reduced enough to cause the sylphon bellows to collapse and close the water valve. As the cold cooling water passes through the condenser, it absorbs heat, thus lowering the pressure and thereby reducing the load of the compressor when it is again started. Figure 97 shows the method of operation of a water-cooled Frigidaire condenser.

Compressor.—The compressor is of the single-acting type having two cylinders, which are driven by an electric motor. The motor drives the compressor by means of one or two V-shaped belts which are nearly noiseless. The compressor runs at a relatively slow speed of about 350 revolutions per minute and is well lubricated by the oil which returns from the cooling unit as already explained. The compressor being of the enclosed type has a stuffing box or seal which is placed around the crankshaft and is sealed with oil to prevent loss of the refrigerant. The latest compressors have a stuffing box known as the "balanced-seal" type. This stuffing box (Fig. 98) derives its name from its design which is such that the internal and external pressures are opposed and nearly balance under operating conditions. Because of this, the amount of wear on the bearing surfaces and the possibility of the stuffing box leaking are greatly reduced, as the springs used exert a pressure of only 30 pounds against the thrust collar on the shaft. In the design of the older seals the spring was placed inside the sylphon bellows while in the balanced seal the spring is outside the bellows. A satisfactory shaft stuffing box to prevent entirely and permanently the leakage of the refrigerant is an essential requirement in any compressor for household refrigeration.

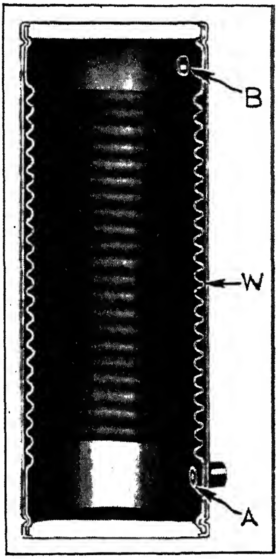


FIG. 97.—Section of wa cooled condenser.

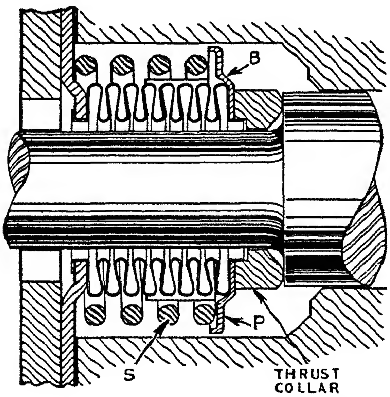


FIG. 98.—Stuffing box of Frigidaire compressor.

The discharge valve of the compressor is of the flapper type shown in Fig. 105, and is clamped in place next to the cylinder by the cylinder head. This

flapper valve consists of a straight reed made of special steel and ground by hand to a perfect fit. The suction valve is located in the head of the piston. The valve and its seat are fitted into a recess in the piston head and are attached to it by means of four screws. A sufficient number of shims are placed under the valve seat to bring the top of the valve assembly flush with the top of the cylinder. The suction valve is of the disk type and is held by a suitable retainer plate which in some valves is locked by merely turning the plate. The suction valve is shown in Figs. 99 and 100.

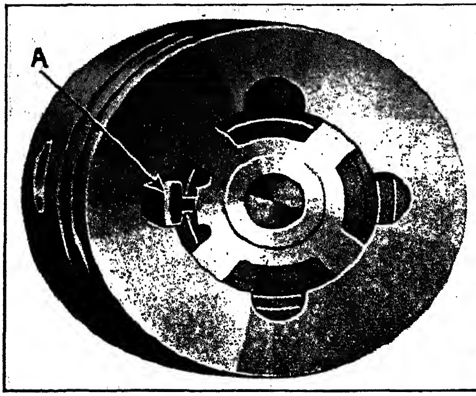


FIG. 99.—Suction valve of small Frigidaire compressor.

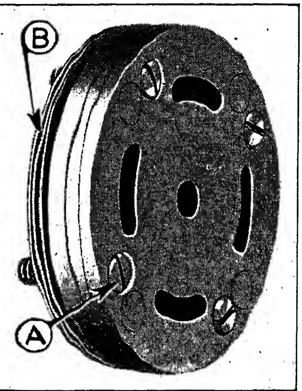


FIG. 100.—Suction valve of commercial Frigidaire compressor.

Spring-rod Suspension.—Springs, as shown at *R* (Fig. 101) are used to support the compressor unit and prevent the transmission of noise and vibration to the base of the machine and then to the building. This suspension is adjustable so as to provide for the leveling of the compressor. Being of the four-point type, the suspension is easily adjusted.

Condenser.—The Frigidaire water-cooled condenser shown in Fig. 97 is of the vertical type, but the horizontal type is also used. It consists of an outer shell pressed over an inner shell in which is formed a spiral groove. The water flows in the spiral groove, and the refrigerant is inside the inner shell. This condenser also serves as a liquid receiver and is generally fastened to the compressor base.

Safety Water Shut-off.—A device called a safety water shut-off is used to stop the motor and compressor when the water is shut

off or is not flowing in the proper amount through the condenser. This safety shut-off device is a part of the automatic water valve

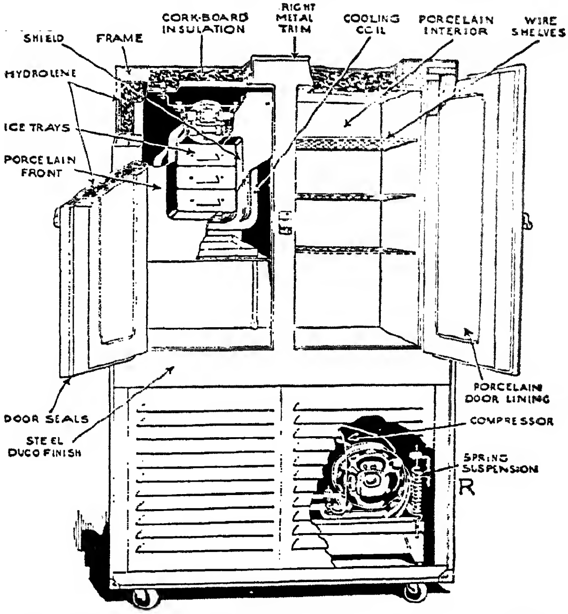


FIG. 101.—Spring suspension of refrigerating machine in base of cabinet.

and is located so that it opens the motor switch when the condenser gage pressure reaches 180 pounds per square inch. It is

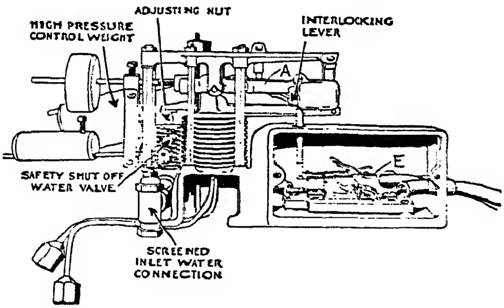


FIG. 101*a*.—Frigidaire safety water shut-off.

operated by attaching a simple interlocking lever between the automatic water-control lever *A* and the motor-switch control *E*, as shown in Fig. 101*a*. When the sylphon bellows of the water

valve is subjected to the above gage pressure, it expands so far that this interlocking lever opens the motor switch, stopping the compressor.

Defrosting.—As all foods stored in a refrigerator cabinet give moisture and the entering air also contains moisture, a coating of frost will accumulate on the surface of the evaporator coils. If the coating of frost accumulates to such an extent as to interfere with the air circulation, the surface of the evaporator should be defrosted. This can easily be done by shutting off the electric

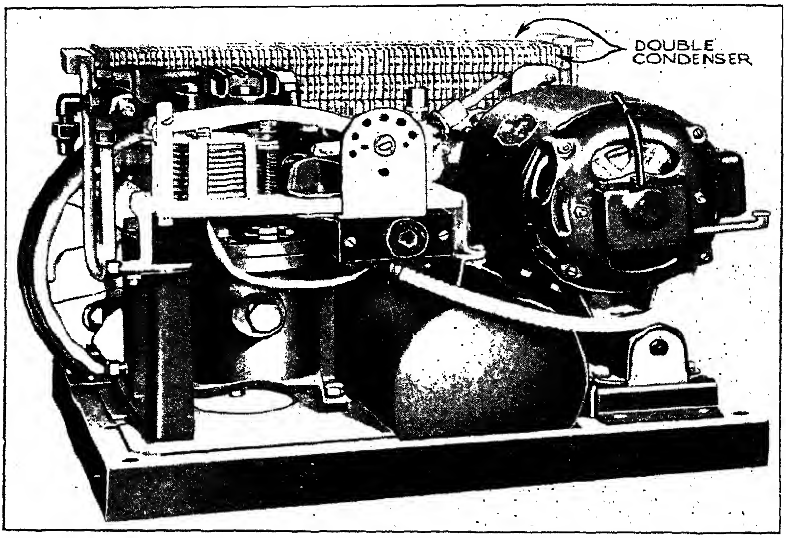


Fig. 102.—Condenser and motor of Frigidaire air-cooled refrigerating unit.

current supply for a period of 12 or more hours. The cabinet will then warm up, and the frost on the surface of the evaporator will melt. The melting of the frost will cool the air within the cabinet, but the temperature in the cabinet will rise slightly. In certain types of equipment where ice cubes are not desired, the evaporating surface may be properly proportioned so that the surface will defrost automatically.

Frigidaire Refrigerating System with Air-cooled Condenser.—The household equipment (Fig. 102) that is generally used in homes and apartments has its condenser cooled by air. The refrigerant now used in this type of equipment is F-12 which as pointed out in Chap. III has unusually good characteristics for this type of

equipment. Figure 103 shows diagrammatically the air-cooled unit consisting of the compressor *C*, the condenser *N*, the cooling unit *A*, the fan *F*, and the liquid receiver *R*. These parts together with an electric motor for driving the compressor and an automatic electric switch are mounted on a steel base, with provision for spring suspension (Fig. 101). A complete air-cooled unit is

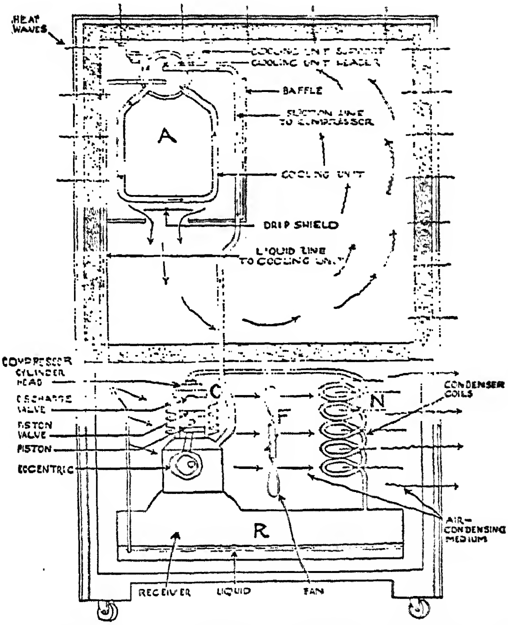


FIG. 103.—Arrangement of Frigidaire refrigerating unit in refrigerator.

shown in Fig. 104. The air for cooling the condenser is drawn in at the bottom and forced out at the back of the cabinet by the fan.

Compressor.—The compressor used in the household equipment is of the single-acting type with twin cylinders. The discharge valve of this compressor is of the simple-flapper type as shown in Fig. 105, while the suction valve is the commonly used disk type both of which are similar to the valves used in compressors intended for water-cooled condensers. The stuffing box or seal is of the balanced-seal type described on page 149. The flywheel on the compressor is provided with fan blades instead of spokes. These fan blades force the air over the cooling

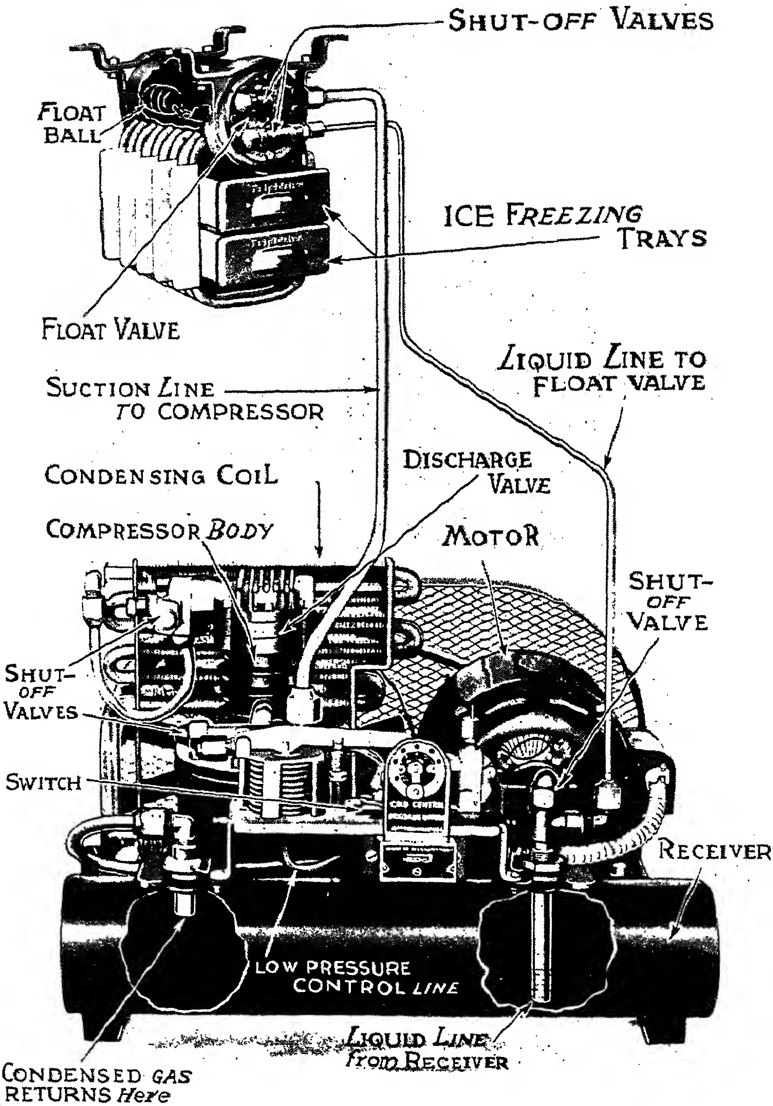


FIG. 104.—Complete air-cooled Frigidaire refrigerating unit.

surface of the condenser, which is located as shown in Fig. 102. A second fan is mounted on the pulley on the motor shaft. A V-shaped belt is used to transmit the power from the electric motor to the pulley on the compressor shaft. The smaller sizes of compressors are operated by $\frac{1}{5}$-horsepower motors, while the larger sizes require $\frac{1}{3}$- to $1\frac{1}{2}$-horsepower motors.

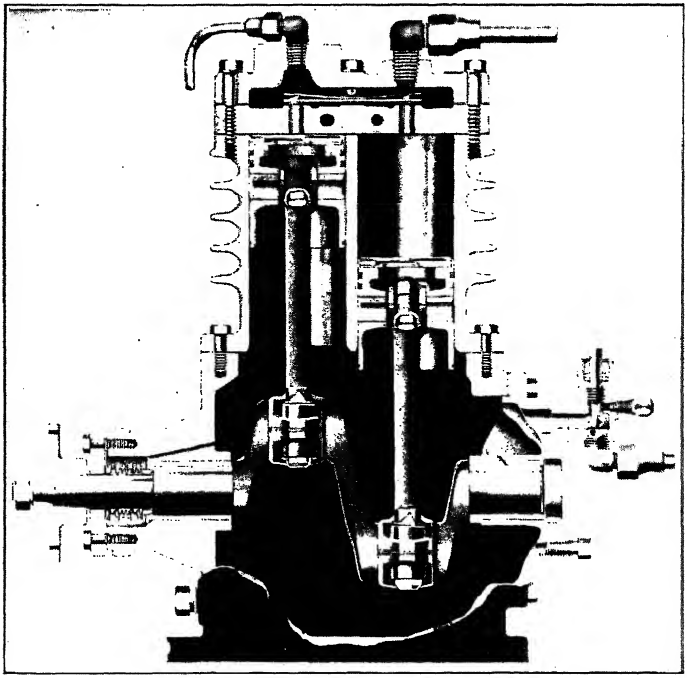


Fig. 105.—Details of Frigidaire compressor (air-cooled).

Control Switch.—The control switch for the air-cooled units is shown in Figs. 106 and 107. It operates by varying the pressure on the low-pressure side. This control switch consists of a sylphon bellows and various linkages with electric contacts. A compensating spring changes the range for both the opening and closing of the switch and generally need not be changed unless other adjusting methods fail. The adjusting nut may be turned up or down which will change the opening point of this switch. The adjustment for changing the point of closing of the

switch is shown. There is also a locknut beneath the cap, while the plunger which extends above the cap passes down and

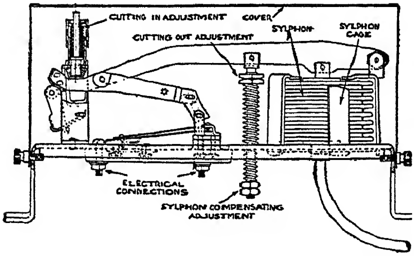


FIG. 106.—Control switch for air-cooled compressor.

rests on the switch arm. A spring inside the cap pushes down on this plunger in accordance with the adjustment of the cap. The

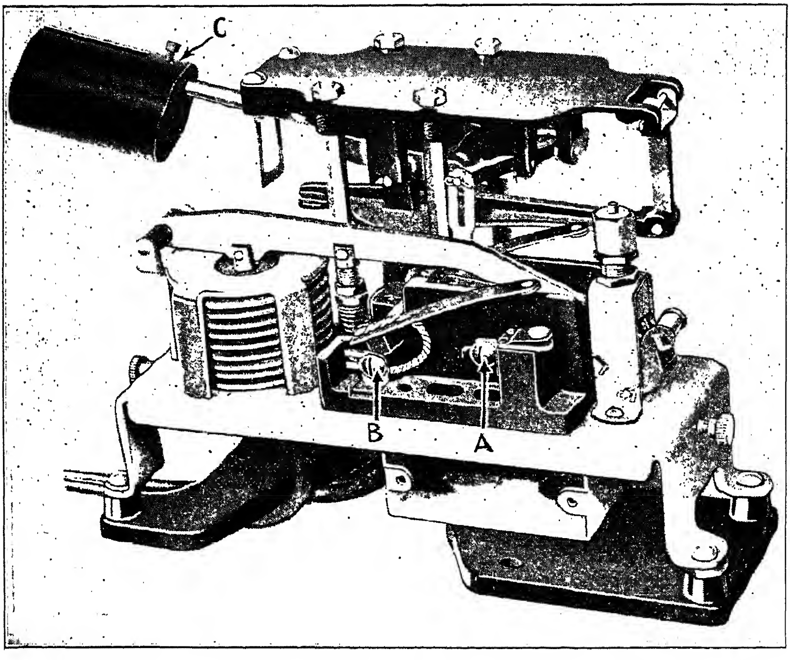


FIG. 107.—Frigidaire switch for large air-cooled compressors.

contact points on this switch are small silver disks. A device to vary the temperature inside the refrigerator cabinet called a

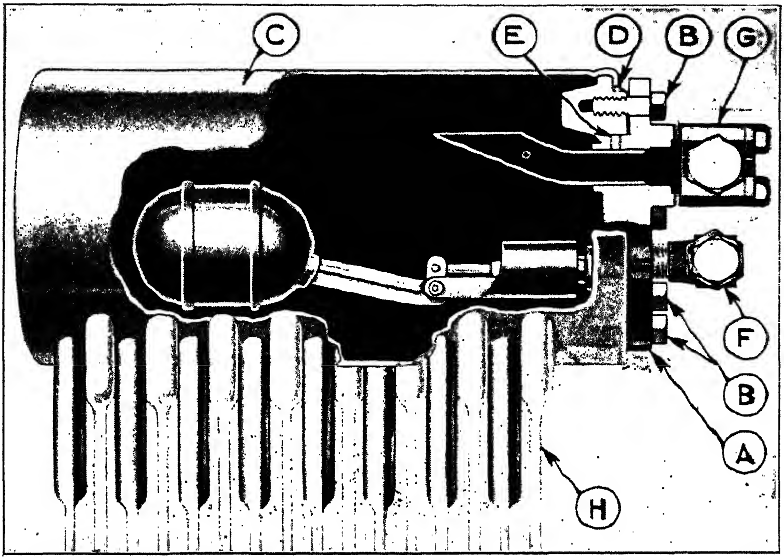


FIG. 108.—Details of Frigidaire cooling unit (evaporator).

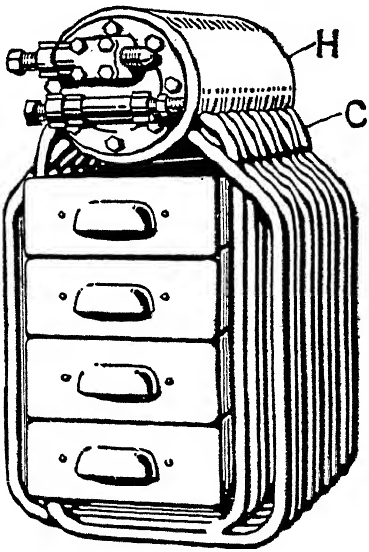


FIG. 109.—Frigidaire cooling coils.

cold control" is attached to the control switch. This permits the user to vary the temperature without changing the adjusting nuts. This is accomplished by the addition of a spring, the tension of which can be increased by turning a dial. Increasing the tension on the cutout arm makes the compressor run longer and thus lowers the pressure and the temperature in the

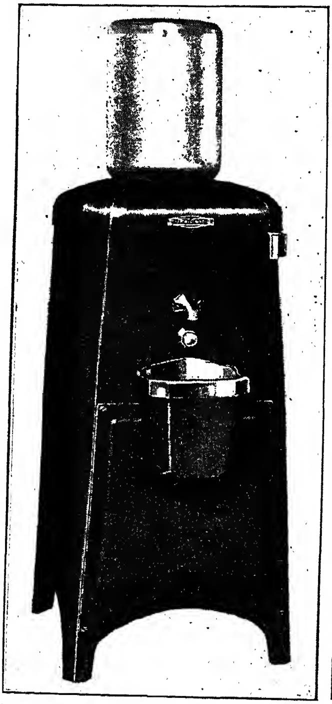


Fig. 110.—Frigidaire water cooler (bottle type).

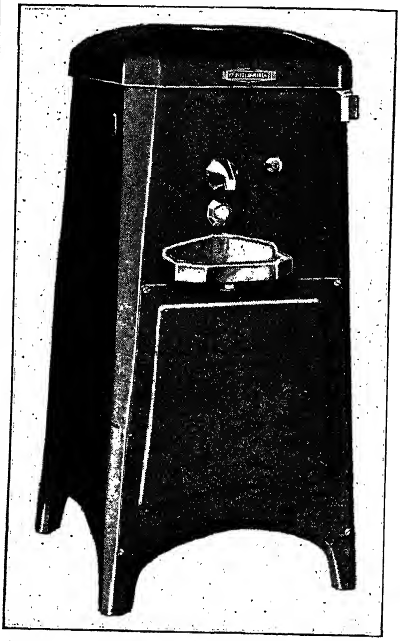


Fig. 111.—Frigidaire water cooler for connection to water-supply piping.

evaporator. It is possible to use a cold control with a refrigerator cabinet which has the compressor and other equipment located at a distance as, for example, in the basement. This is accomplished by extending the Bouden wire (flexible cable) from the refrigerator to the compressor switch.

Cooling Unit.—The Frigidaire cooling units are made in many sizes and shapes to meet the different refrigeration requirements. Although made in many shapes the fundamental principle

is the same in all of them. A typical cooling unit made for a household refrigerator is shown in Fig. 108. This cooling unit operates on the flooded principle (p. 63). The float chamber, shown at the top of the figure, contains the float valve which is shown in detail in Fig. 93. This float valve controls the supply of liquid refrigerant to the cooling unit. The cooling coils or tubes *C* (Fig. 109) are made of copper and the tubes are silver-soldered to the bottom of the float chamber *H*. Ice trays are placed within the coils. The inside surface of the cooling coils, therefore, cools the ice trays, and the outside surface of the coils chills the air which circulates through the food compartment.

A typical Frigidaire cabinet is shown in Fig. 101. The exterior is made of sheet steel finished with white acid-resisting porcelain. Rock wool of a thickness equivalent to about 2½ to 3 inches of cork board is used for insulating the walls of the cabinet. The interior is finished in white porcelain, while the hardware is satin-chrome finish. The doors are insulated with lighter weight heat insulators, such as dry zero (p. 368).

TABLE *Va*.—COMPRESSORS AND CONDENSING UNITS MADE BY THE FRIGIDAIRE CORPORATION
Refrigerant, F-12

Size of compressor, inches	Number of cylinders	Compressor speed, revolutions per minute	Motor horsepower	Capacity* B.t.u. per 14 hours	Type of condenser
1½ × 1 7/16	2	255 350	1/5 1/4	8,350 11,950	Air Air
1 11/16 × 2⅜	2	255	1/3 1/3	24,300 26,300	Air Water
1⅞ × 2⅜	2	380	1/2 3/4	37,200 37,200	Water Air
2⅛ × 2⅜	2	400	3/4 1	53,200 53,200	Water Air
2⅛ × 3¼	2	360	1 1½	74,400 74,400	Water Air

* Capacity ratings are based upon air or water supplied to the condenser at 80° F., a suction pressure of 12 pounds per square inch gage, and a 20° F. cooling-coil temperature for 14 hours' operation.

Interesting recent applications of Frigidaire refrigerating units are in water coolers using bottled water (Fig. 110) and those having water-pipe connections with the city water supply (Fig. 111).

Compressor and Condenser Data for Frigidaire Equipment.—Table V*a* gives some data on compressor sizes and capacities for various speeds. The sizes vary from household and ice-cream cabinet into the smaller commercial compressors for both air- and

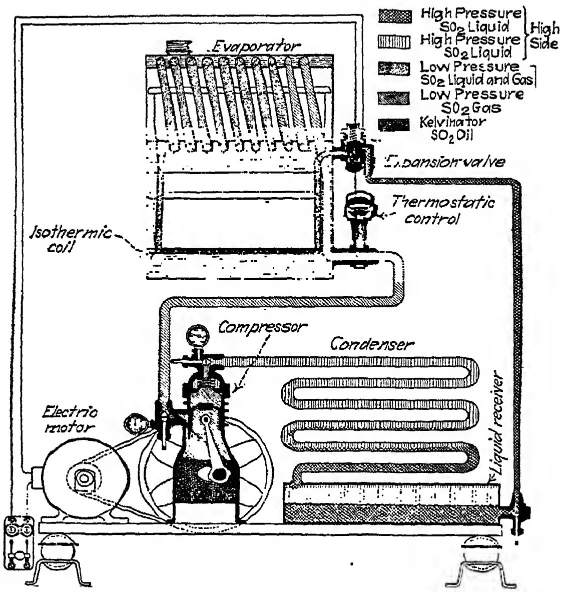


FIG. 112.—Kelvinator household refrigerating unit (dry system).

water-cooled condensers. In making comparisons, the suction pressure and condenser-cooling medium temperature should be noted.

Kelvinator Refrigerating Systems.—The Kelvinator[1] Corporation manufactures several types of air- and water-cooled refrigerating equipment. The household or domestic units are air-cooled, while the commercial equipment may be either air- or water-cooled, the maximum capacity being about 1¼ tons of refrigeration. The refrigerant used in most localities is sulphur dioxide

[1] Derived from the name of Lord Kelvin, noted physicist, who did a great deal of experimental work in thermodynamics.

but, where there are restrictions upon the use of this refrigerant, F-12 (p. 86) is used. In the ice-cream and frozen-food cabinets, methyl chloride with a slight trace of sulphur dioxide is used. The sulphur dioxide is used in combination for detecting leaks.

The refrigerator cabinets are insulated with a $2\frac{1}{2}$-inch thickness of a material similar to celotex (p. 367) and are provided

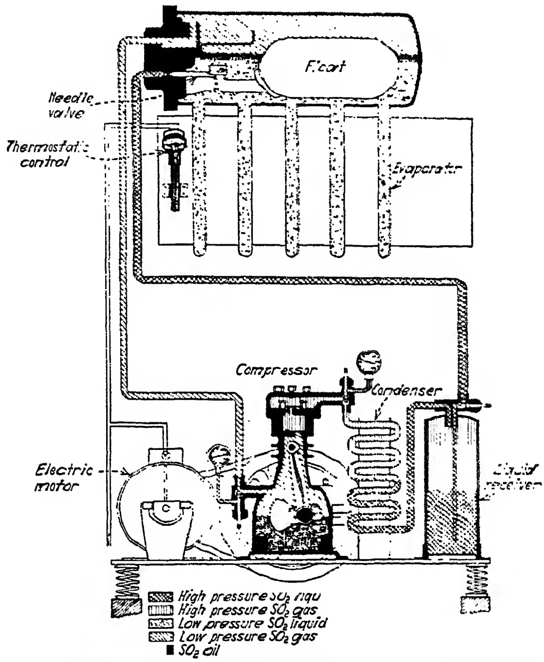


Fig. 113.—Kelvinator household refrigerating unit (flooded system).

with a temperature-control device attached to the cooling unit with an adjustment for five rates of freezing.

Kelvinator Dry and Flooded Systems.—The *dry system* is shown in Fig. 112 and is made up of the usual equipment; namely, compressor, cooling unit, condenser, liquid receiver, expansion valve, and temperature control. This system is commonly used with refrigerator cabinets located in private houses and single apartments, as the quantity of refrigerant is small thus reducing its cost. The expansion valve is of the automatic type (p. 15).

The *flooded system* (p. 63), shown in Fig. 113, differs chiefly from the dry system in the type of cooling unit used. The flooded system with a float valve (p. 146) is used in installations where one or more cooling units are to be refrigerated by one compressor.

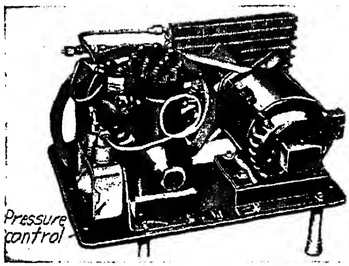


Fig. —Kelvinator household frigerating unit.

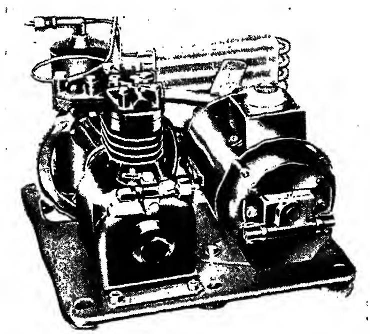


Fig. 115.—Kelvinator commercial refrigerating unit.

Compressor.—The compressors are of the single-acting type having one, two, or four cylinders depending on the capacity required (Figs. 114 and 115). The pistons of the medium-size compressors slide in steel sleeves. The type of valves used varies with the different types of compressors. In some compressors the suction valves are in the piston head (Fig. 116) and are of the poppet, reed, and disk-ring types (p. 150). In several designs of the larger compressors, the *suction* valve which is of the poppet type is located in the *discharge* valve plate. In this design the servicing of the suction valve is greatly facilitated as the discharge valve plate may easily be removed. The discharge valves of the reed and disk type are in a plate, and the valves are protected from damage by means of a safety spring which allows the valve casing to lift when subjected to excessive pressures. The flywheel has a V-shaped groove for the belt drive and a steel rim and spokes with fan-like blades. There is also a fan on the pulley of the motor. The larger heavy-duty commercial compressors are built

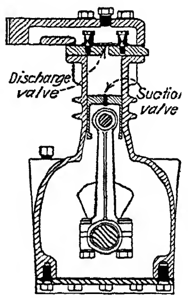


Fig. 116.—Kelvinator compressor with suction valves in piston head.

with either water or sulphur-dioxide-cooled (Fig. 117) cylinder heads, permitting these compressors to handle heavy refrigerating

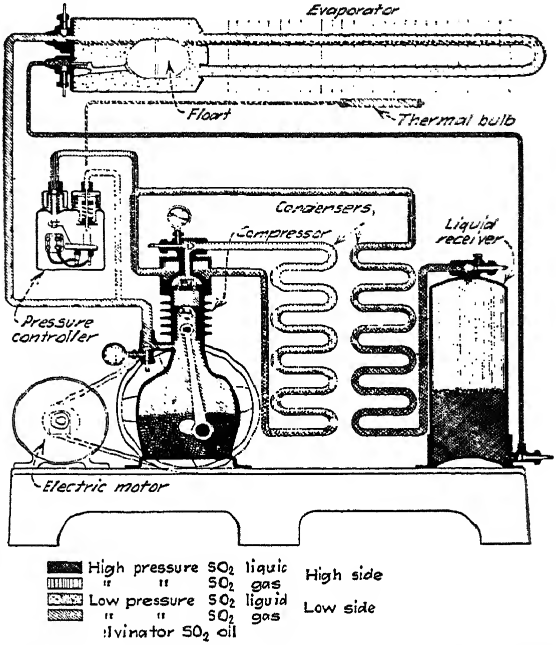


FIG. 117.—Kelvinator heavy-duty compressor cooled with sulphur dioxide.

loads. The compressor seal is of the balanced type (p. 149) and is shown in Fig. 118. It consists of a sylphon bellows having a self-lubricating metal collar which is pressed against the crankshaft shoulder by a spring. In the smaller domestic models the compressor running time for a room temperature of 100° F. is about 50 per cent, while for the larger models the machine operates about 35 per cent of the time. The average number of operations per day is about twelve to fifteen. Data regarding the Kelvinator condensing units are shown in Table Vb.

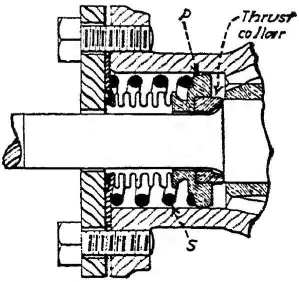


FIG. 118.—Stuffing box for Kelvinator compressor.

Liquid Receivers.—Liquid receivers are made both vertical and horizontal, with preference for the vertical type for the following

reasons: (1) No exact amount of liquid refrigerant is required in the vertical type; (2) non-condensable gases may be easily purged; (3) less chance of dirt entering the liquid line; (4) greater effective capacity for a given occupied space.

Expansion Valve.—In this system the expansion valve may be of the automatic type, as shown in Fig. 119. In this type the

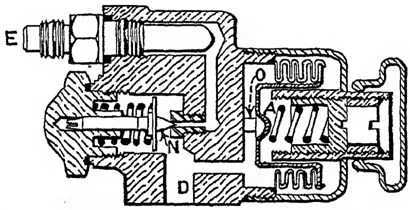


FIG. 119.—Kelvinator automatic expansion valve.

adjusting spring is separated from contact with the low-pressure vapor by means of a sylphon bellows. The bellows is actuated by the adjusting spring *A*, and the pressure of the vapor moves the operating pin *O* which moves the needle valve *N*. A high pressure in the evaporator closes the needle valve *N*, as this pressure overcomes the action of the adjusting spring. When the pressure has been reduced by the operation of the compressor, the adjusting spring opens the needle valve and allows more refrigerant to enter the cooling unit until the increasing pressure closes the valve.

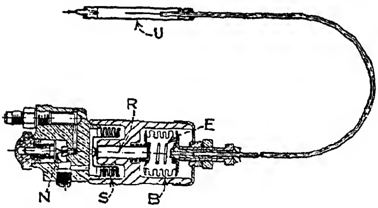


FIG. 120.—Kelvinator thermostatic expansion valve.

Another type of expansion valve used with the dry system is the *thermostatic* expansion valve, shown in Fig. 120, which has a bulb *U* containing sulphur dioxide attached by tubing to the operating bellows *B*. It is clamped to the suction line where it leaves the cooling unit, and the pressure within the bulb changes with the temperature of the vapor in the suction line. The rising pressure in the bulb expands the bellows *B*, moving the rod *R* which forces the needle valve *N* to open. A decrease in the suction-gas temperature reduces the pressure in the tube and bellows, causing the rod *R* to close the needle valve. With certain conditions, when the temperature of the vapor leaving

the cooling unit is higher than it should be, the valve remains open until the pressure rises enough to operate the second bellows S and close the needle valve N. This will cause the pressure to fall, lowering the temperature in the cooling unit.

121.—Kelvinator unit.

Float Valve.—In the flooded system the float type of expansion valve is used. This valve is shown in Fig. 113 and is located on the low-pressure side of the system.

Cooling Unit.—The cooling unit made for domestic cabinets consists of a tinned copper shell having a rectangular cross-section. The joints are soldered. The tank of the cooling unit contains several "sleeves" for ice-freezing trays and a tray for frozen foods. The tank is filled with a 35 per cent solution of alcohol. The alcohol solution acts as a stabilizer for the cabinet as it stores up considerable refrigeration which reduces the number of operations of the compressor. Copper tubing is coiled around the "sleeves" for removing the heat. These coils may be seen in Fig. 121 which also shows the "pancake" coil soldered to the bottom of the lower ice-tray sleeve. Another view of this pancake coil is shown in Fig. 112, the purpose of which is to freeze rapidly ice cubes without the necessity of setting a manually controlled device. Where there are several cooling units operated in multiple, the flooded type of cooling unit with float valves, as shown in Fig. 113, is used. Fins are attached to the cooling coils for quickly extracting the heat from the air in the cabinet. Several sleeves are provided, the number depending on the size of cooling coil.

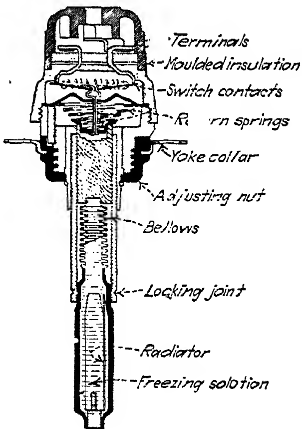


FIG. 122.—Details of thermostatic electric control.

Temperature Control.—The starting and stopping of the electric motor are accomplished by the use of a thermostat shown in Fig. 122. This thermostat in the dry system is clamped to the suction line, as shown in Fig. 112, near the cooling unit, while in the flooded system it is clamped as shown in Fig. 113 to the fins. The thermostat opens one side of the line with a quick opening of the contacts. As shown in Fig. 122, the lower end of the thermostat contains a freezing solution which on freezing expands against the sylphon bellows, forcing the "return" spring to be compressed and moving the spindle upward, thus causing the toggle plate to buckle upward to open the electric contacts. Likewise, when the temperature of the end of the thermostat rises, the freezing solution melts allowing the return spring to move the spindle downward, causing the toggle plate to buckle downward to close the electric contacts.

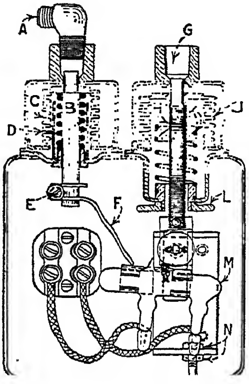


FIG. 123.—Kelvinator temperature-pressure controller.

Another type of electric control used with Kelvinator commercial equipment is shown in Fig. 123. This electric control is known as the "pressure control," which is an electric-mercury switch operated by two separate sylphon bellows, one being connected to the low-pressure side while the other is connected to the high-pressure side. The bellows on the low-pressure side is usually connected to the crankcase and the one on the high-pressure side to the cylinder head of the compressor.

The low-pressure controller is operated by the pressure of the vapor in the suction line, and it is operated, therefore, indirectly by the temperature of the cooling unit. The purpose of the high-pressure bellows is to open the switch when the pressure on the high-pressure side becomes too high because of unusual operating conditions.

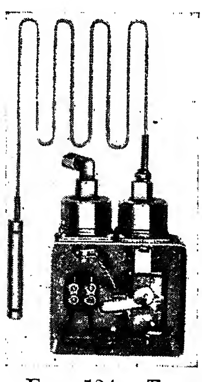


FIG. 124.—Temperature-pressure control device.

Still another control is that known as the "temperature-pressure control," shown in Fig. 124, which is a type of mercury-

electric switch operated by means of two separate sylphon bellows. One is operated by the pressure on the high-pressure side so as to open the electric circuit when the pressure on that side becomes too high, while the other is connected to a temperature-control bulb. The high-pressure bellows is generally connected to the cylinder head of the compressor.

TABLE V*b*.—CONDENSING UNITS MADE BY THE KELVINATOR SALES CORPORATION

Size of compressor, inches	Number of cylinders	Compressor speed, revolutions per minute	Motor horsepower	Capacity,* B.t.u. per 14 hours	Type of condenser
$1\frac{1}{4} \times 1\frac{1}{2}$	1	525	$\frac{1}{8}$	8,000	Air—radiator
$1\frac{13}{16} \times 1\frac{1}{2}$	1	350 410	$\frac{1}{6}$	9,000 11,300	Air—radiator
$1\frac{1}{4} \times 1\frac{1}{2}$	2	525	$\frac{1}{4}$	14,850	Air—radiator
$1\frac{13}{16} \times 1\frac{1}{2}$	2	340	$\frac{1}{4}$	16,800	Air—radiator
$1\frac{13}{16} \times 1\frac{1}{2}$	2	430	$\frac{1}{3}$	21,200	Air—radiator Water—dual spiral tube
$1\frac{13}{16} \times 2\frac{1}{2}$	2	310 360	$\frac{1}{3}$	33,600 40,600	Air—radiator Water—dual spiral tube
$1\frac{13}{16} \times 2\frac{1}{2}$	2	450 500	$\frac{1}{2}$	50,000 53,200	Air—radiator Water—dual spiral tube
$2\frac{1}{4} \times 3\frac{1}{2}$	2	280 330	$\frac{3}{4}$	71,000 81,200	Air—radiator Water—dual spiral tube
$2\frac{1}{4} \times 3\frac{1}{2}$	2	400 490	1	96,000 110,600	Air—radiator Water—dual spiral tube
$2\frac{1}{4} \times 3\frac{1}{2}$	4	330 490	2 3	162,400 221,200	Water—dual spiral tube

* Capacity ratings are based upon air or water to condenser at 80° F. and a suction pressure of 0 pounds per square inch gage for 14 hours' operation.

The temperature-control bulb is filled with sulphur dioxide, the pressure in the bulb changing with the temperature in the refrigerator and operating the temperature bellows so as to start and stop the compressor at the proper temperatures.

Condensers.—The air-cooled condenser used with household or domestic refrigerators is of the radiator type (p. 154) consisting of copper tubing to which fins are soldered. Air supplied by means of the fan on the motor pulley is driven through the radiator. The water-cooled condenser consists of a vertical steel shell which serves also as a liquid receiver (p. 14). It contains a spiral water tube.

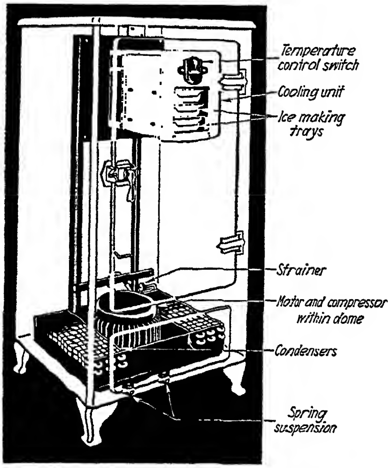


FIG. 125.—Phantom view of Servel compression type of household refrigerator.

Servel Compression Household Refrigerator.—The refrigerating unit manufactured by Servel Sales, Inc., for household use consists of a hermetically sealed unit located in the bottom of the refrigerator cabinet, condensers at each side, as shown in Fig. 125, the cooling unit being at the top of the cabinet. The refrigerating unit is shown more in detail in Fig. 126. The intake for the liquid refrigerant is marked. The discharge line is on the side of the unit which is not shown in the figure. The essential parts of the refrigerating unit consist of an electric motor supported on a vertical shaft which is directly connected without gears or belts to a reciprocating compressor of which the cylinder is in a horizontal position. The refrigerating unit is placed with an air-tight seal under a hat-shaped dome. The crosshead of the compressor serves as a mechanically operated intake valve through which the vapor of the refrigerant is delivered into the cylinder of the compressor. The piston is stationary, and its cylinder has reciprocating motion. When the vapor of the refrigerant is to be discharged into the compressor, the piston uncovers a small slot in the cylinder wall just before the end of the stroke. The compressed vapor is discharged through the hollow piston into the condensers.

The details of the compressor and oil pump are shown in Fig. 127.

When the refrigerating unit is started by closing the electric switch, the current flows first through the starting winding of the motor (Fig. 126). This current in the starting winding sets up a magnetic field which is strong enough to turn the motor without a load and serves also to pull down the "unloading" valve from its

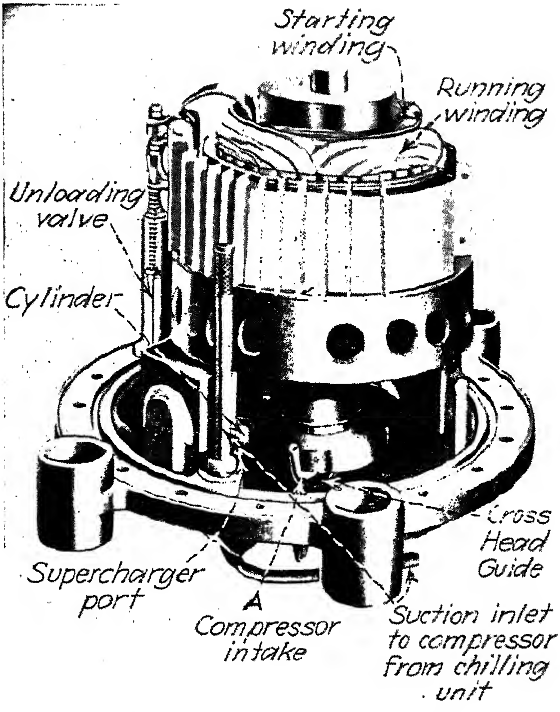


FIG. 126.—Servel compression refrigerator unit.

seat, thus permitting the motor to operate the oil pump without a load. There is no load on the compressor at this time as the unloading valve equalizes the pressure on the two sides of the valve. When the electric motor reaches its normal speed, the current is shut off from the magnetic field and the unloading valve returns to its closed position. In this position of the unloading valve, the compressor becomes effective for taking in,

compressing, and discharging compressed vapor into the condensers, the vapor entering the suction pipe of the compressor from the hat-shaped dome.

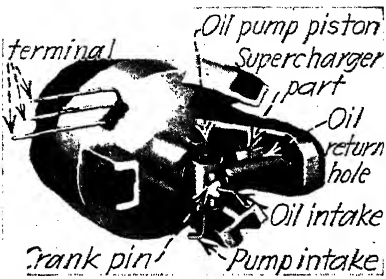


FIG. 127.—Servel compressor and oil pump.

Thermostatic Control Switch. The control switch and thermostat are located at the back of the frame supporting the cooling unit. This thermostatically operated switch controls automatically the starting and stopping of the compressor, its operation being governed by the changes of temperature in the cooling unit. Included with this regulating switch is an emergency overload switch to protect the electric motor against damage from low voltage or overload.

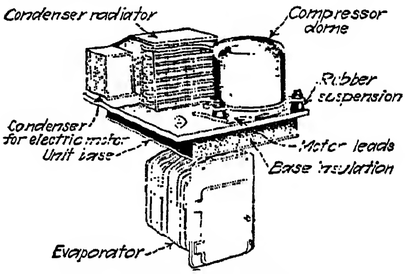


FIG. 128.—Majestic refrigerating unit.

Majestic Refrigerating Unit.—One of the latest models to be introduced into the refrigerating field is the Majestic. This household unit is constructed with a hermetically sealed compressor (the system being of the compression type), which does away with the stuffing box and belt, as the compressor is directly driven by the motor. As shown in Fig. 128, the compressor dome, electric condenser, and condenser are all located on the upper side while the cooling unit is attached to the lower side of the cover. This cover may be replaced when necessary as it simply sets in position. A diagram of the complete unit is shown in Fig. 129.

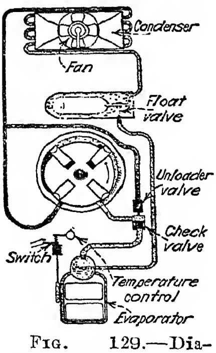


FIG. 129.—Diagram showing operation of Majestic unit.

Compressor.—The compressor is of the rotary type, with a vertical shaft, driven by a $\frac{1}{6}$-horsepower squirrel-cage[1] motor of

[1] See MOYER and WOSTREL, "Industrial Electricity and Wiring," McGraw-Hill Book Company, Inc., New York.

the induction type at 1,725 revolutions per minute. The dome is suspended on rubber to eliminate the transmission of noise. In addition to compressing the sulphur dioxide, the rotary compressor acts as an oil pump. This is accomplished by using the space back of the compressor blades. As shown in Fig. 130, the oil is drawn in through a groove in the bottom plate. A little later in the revolution, the blades force the oil up through a hole in the motor shaft, and by means of other suitable holes and grooves the motor bearings are lubricated. There are four of these pulses of oil discharged per revolution. The supply is about 1 pint of oil per minute. Vibration is almost eliminated as the hub which holds the four blades is concentric with the driving shaft. The rotor of the motor is also balanced. The power input to the motor is about 205 watts. A cross-section of the compressor dome is shown in Fig. 130.

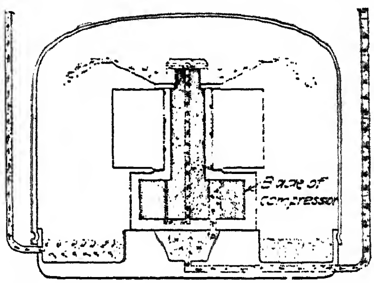


Fig. 130.—Oiling system in compressor dome.

Condenser.—The condenser is the radiator type and is made of copper tubing with fins. An electric 7-inch fan consuming about 25 watts of power circulates about 30 to 100 cubic feet of air per minute through the condenser. The air is drawn upward from the floor. All connections are silver-soldered which otherwise often fail when subjected to high temperatures. The condenser is located at the top of the cabinet.

Evaporator and Float Valve.—A flooded type of evaporator is used having the float chamber located on the cover above the evaporator. This float expansion valve is the "high-side" type, controlling the liquid level in the float chamber as more refrigerant is supplied from the condenser. The evaporator is made of copper tubing formed in loops within which are placed the ice-tray sleeves. The evaporators are designed in sizes for producing from 4¼ to 8½ pounds of ice. The evaporator and the condenser are shown in Figs. 128 and 129.

Temperature Control.—A thermostatic bulb is clamped to the evaporator for starting and stopping the electric motor. The bulb pressure actuates a sylphon bellows (p. 164) to which a temperature regulator is attached. A knob on the outside of

the cabinet adjusts the temperature through a temperature range of 38 to 45° F.

Overload Trip Switch.—The trip automatically takes care of line surges due to lightning without the need of replacing burned-out fuses. It automatically starts the motor three times to see if the trouble has been cleared, after which it shuts off the current and locks the switch, and lights a green pilot light which serves as an indicator.

Unloader and Check Valve.—The check valve being located in the suction line prevents the return of compressed vapor into the evaporator when the compressor is stopped. It is self-operating, being merely a weighted disk. The *unloader* is for the purpose of equalizing the pressure in the compressor which will reduce the

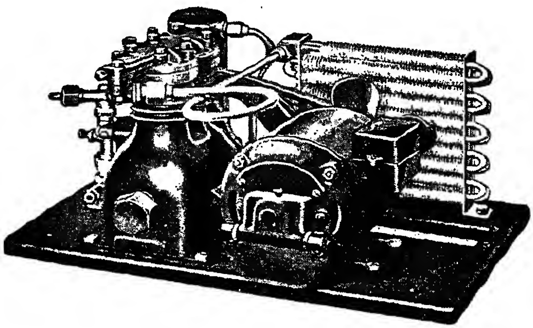


FIG. 131.—Copeland household refrigerating unit.

starting torque. This valve is located in the compressor dome and is actuated by the change in pressure of the refrigerant when the compressor starts.

Cabinet.—The cabinet is made of steel, the exterior being finished with pyroxylin lacquer, while the interior is finished with kaolin porcelain on seamless steel. The hardware, such as door latches and hinges, is made of bronze and chrome composition with satin finish. The cabinet is insulated with 3 inches of "dry zero" insulation (p. 368).

Copeland Refrigerating Systems.—The Copeland Sales Company manufactures both domestic and small commercial refrigerating equipments. The refrigerant used in the domestic refrigerators is isobutane, while in the commercial equipments methyl chloride is used. The dry cycle of compression is generally used, the commercial equipment having thermostatic-expansion valves.

Compressor.—The compressor is either a single or twin reciprocating type, driven by an electric motor, as shown in Fig. 131. These compressors are operated at different speeds, 360, 390,

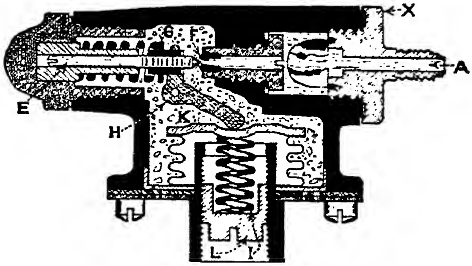


FIG. 132.—Copeland expansion valve.

and 440, depending upon the refrigerating capacity needed. The discharge and suction valves are of the disk type, and the usual safety spring is used with the discharge-valve plate. The compressor is driven by means of a "V"-shaped belt and notched flywheel. The seal is the sylphon-bellows type, the sealing ring being pressed against the crankshaft by a spring. The motor horsepower for domestic units is one-sixth or one-fourth.

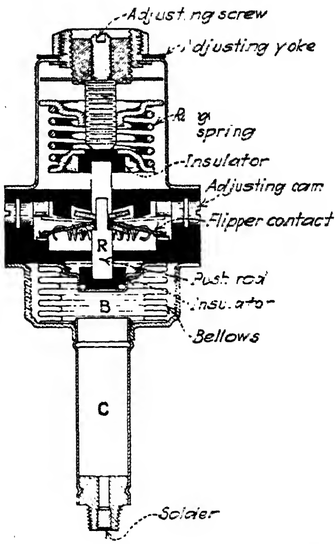


FIG. 133.—Copeland temperature-control device.

Condenser.—The domestic units are cooled by air and are of the radiator type made of a copper coil covered with fins. The air for cooling is supplied by a fan on the electric motor.

Chilling Unit.—The evaporator consists of a tank filled with a 40 per cent solution of denatured alcohol, in which is placed a copper tubing formed to fit around the ice trays. The expansion valve is on the top of the tank.

Expansion Valve.—The expansion valve is shown in Fig. 132. The liquid refrigerant enters the valve at *A* passing first through the strainer, and then through the needle-valve seat *F* to the

outlet (not shown in the figure) located at the bottom of the valve. The spring for opening this valve is shown at *I*, being adjusted by the screw *L*. The suction pressure acts against a sylphon bellows instead of a diaphragm, for moving the needle-valve lever *H* for opening or closing the needle valve *E*.

Temperature Control.—The temperature control located at the back of the chilling unit is shown in Fig. 133. The lower end is filled with isobutane. The pressure in the chamber *C* at this end changes with the temperature and operates the sylphon bellows *B*, which forces the push rod *R* to move causing the "flipper" contacts to open or close. An adjusting screw is shown at the top by which it is possible to change the temperature of the cabinet. The control is provided with a yoke and rod by which the temperature may be set by hand. This device is shown in Fig. 134. The "cutting in" and "cutting out" points of this control may be changed by moving the pair of differential cams which when revolved will change the position of the contacts.

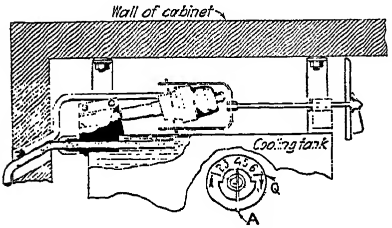


FIG. 134.—Manual temperature control.

Liquid Receiver.—The liquid receiver is a vertical shell with the inlet at the top and the outlet at the bottom.

Cabinets.—The cabinets are either finished on the exterior with white lacquer on steel or with porcelain. The interior is lined with porcelain. The insulation is 2½ or 3 inches of cork board.

Commercial Equipment.—The commercial equipment resembles the household refrigerating units except that the former is of larger capacity. The motor horsepower varies from one-fourth to one and one-half. The condenser may be either air- or water-cooled. The air-cooled condenser is the radiator type, while the water-cooled condenser is made of a vertical shell containing a three-row coil through which the water passes. The condensed refrigerant collects at the bottom of the shell, thus serving as the liquid receiver.

Calcium-chloride Drier.—The presence of moisture often leads to expansion valve trouble which may be prevented by the use of a drier. It should be used on new installations or whenever drying is necessary. The principle of the drier is based on the excep-

tionally strong affinity of calcium chloride for water. The drier consists of a tube filled with chemically pure calcium chloride, which is held in place by a wad of glass wool packed in each end of the tube. The drier should be allowed to remain in the liquid lime approximately 2 hours. The charge of calcium chloride should be replaced after the moisture has been extracted from the refrigerant, as the quantity of moisture that can be absorbed by the calcium chloride is limited. The drier is shown in Fig. 135.

The *expansion valve* used in multiple installations resembles quite closely the valve shown in Fig. 136, which is used with the domestic equipment. This valve has been changed by removing the adjusting spring and nut and adding a sylphon bellows and push rod. The sylphon bellows is connected to a tube having at the end a bulb filled with a volatile liquid. The bulb is con-

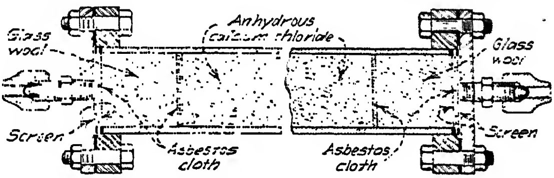


FIG. 135.—Copeland drier.

nected to the suction line. With these changes the expansion valve is made into a typical *thermostatic expansion valve.*

The Penn pressure control (Fig. 136) operates by means of a sylphon bellows connected to the crankcase chamber of the compressor and is, therefore, functioning with changes in the suction vapor pressure, and another sylphon bellows to stop the motor if the discharge pressure becomes too high, the latter being connected to the piping containing the liquid refrigerant.

Rotary Type of Compressor for Household Refrigeration.—A household refrigerating machine which has a gear type of rotary compressor is shown in Fig. 137 and a rotary-pump type in Fig. 138. The refrigerant in this type is usually *ethyl* or *methyl chloride.*

An interesting compressor of the rotary type is part of the equipment of the Williams household refrigerating system made by the Simplex Refrigerating Corporation of Brooklyn, N. Y. A sectional drawing of this rotary compressor is shown in Fig. 139. *Ethyl chloride* is used as the refrigerant. It is stated that this compressor has a volumetric efficiency of about 82 per cent

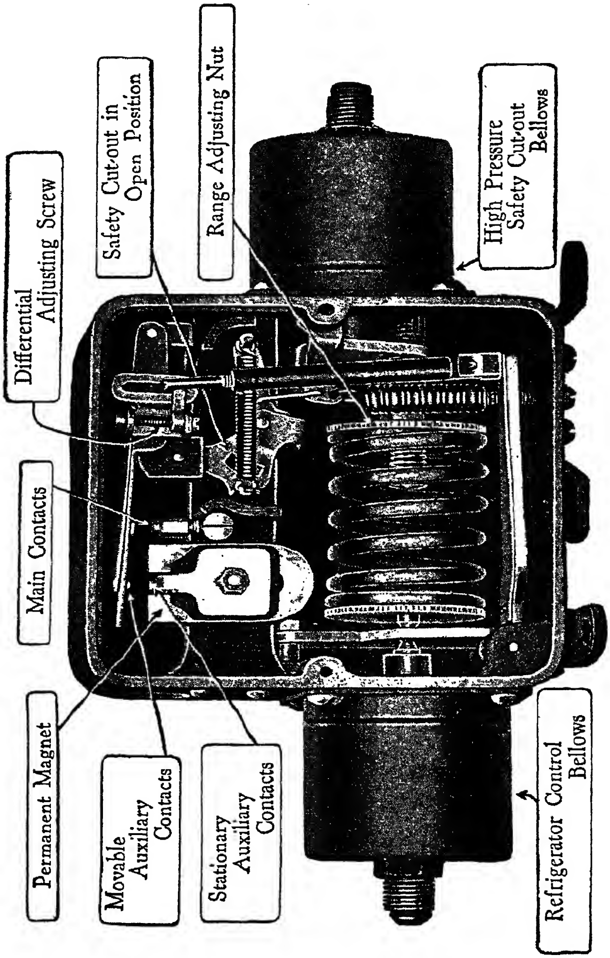


FIG. 136.—Penn control switch.

and that its mechanical efficiency compares favorably with that of the reciprocating types.

Norge Household Refrigerating System.—The compressor used in the Norge equipment is of the rotary type. The compressing element is an eccentrically driven roller moving in a closed cylinder, as shown in Figs. 140, 141, 142, 143 and 144.

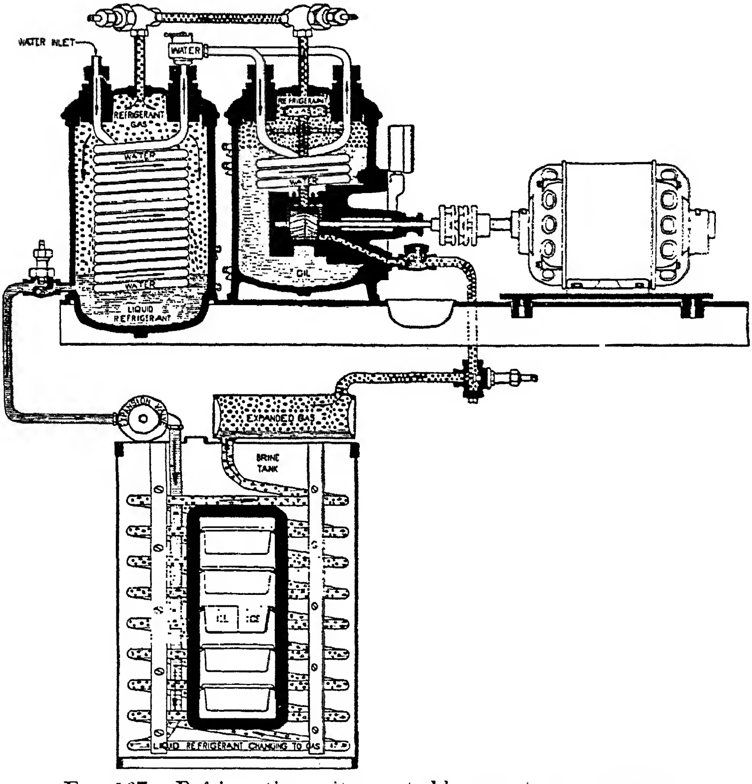


Fig. 137.—Refrigerating unit operated by gear-type compressor.

The intake and the discharge passages are separated by a blade which always maintains contact with the outside surface of the roller. The end surfaces of the cylinder are sealed by a film of oil to prevent the leakage of the vapor of the refrigerant. The roller does not revolve at the same speed as the shaft driving it but simply rolls slowly around the eccentric E on the shaft of the compressor. The reciprocating movement of the blade

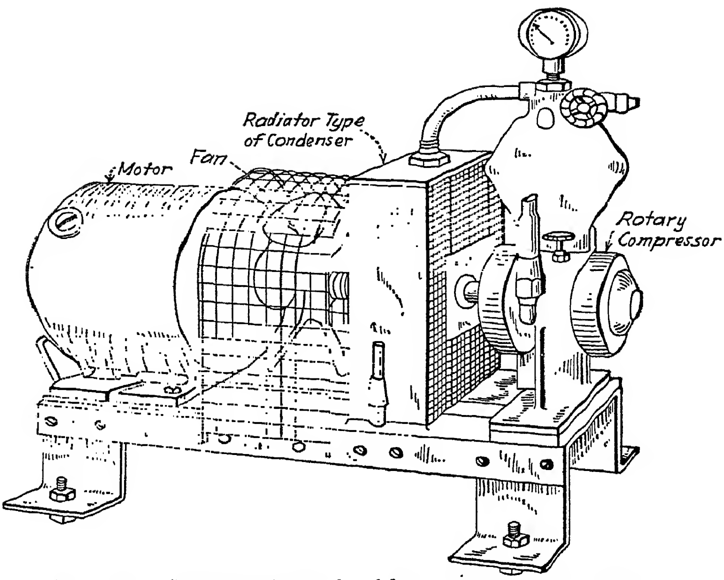


FIG. 138.—Refrigerating unit with rotary-pump compressor

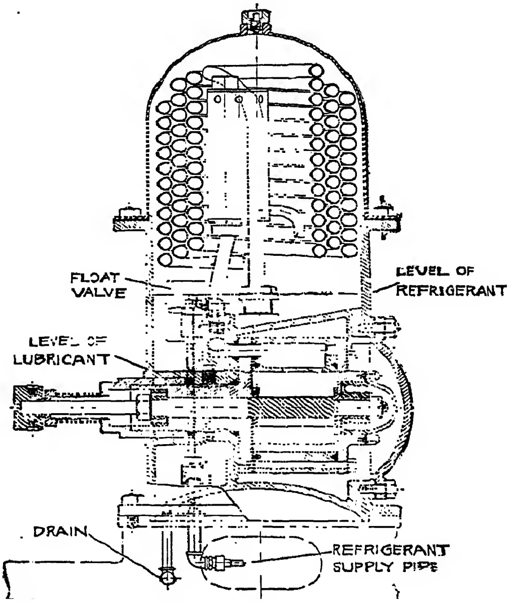
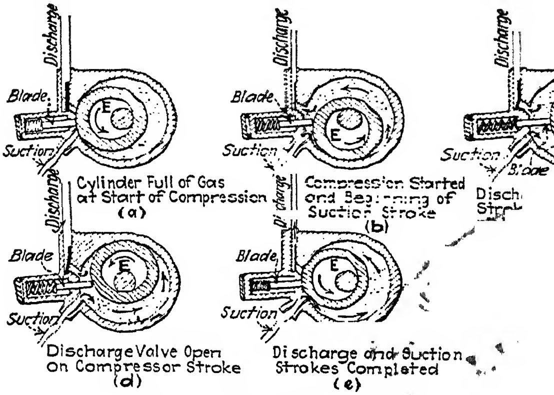


FIG. 139.—Refrigerating unit operating in oil bath.

plainly marked in the figures admits at the proper time the refrigerant vapor to be compressed and after compression discharges it, as shown in *d* (Fig. 143).

The moving parts in this compressor are again shown in Fig. 145. There is a film of oil at all times between the moving



Figs. 140–144.—Diagrams of Norge rotary compressor.

surfaces, so that wear, especially at the ends of the roller, is reduced to a minimum. There is no danger of drawing air or moisture into the system through the oil seal, as the pressure on the oil is always outward.

The condenser, liquid receiver, and compressor, as applied in the Norge system, are shown in Fig. 146. The float valve in the evaporator is in the shape of a pan, as shown in Fig. 147. The required temperature is maintained in the refrigerator cabinet by the thermostat, similar to the one illustrated in Fig. 122. The refrigerant used in this system is sulphur dioxide.

Fig. 145.—Norge compressor.

The oil circulates through the system with the refrigerant. A certain amount is constantly passing through the discharge valve into the condenser with the compressed refrigerant vapor. At the high pressure maintained in the condenser, the oil mixes with the liquid sulphur dioxide, and this mixture flows first into the liquid receiver and then into the evaporator. During the vaporization of the liquid sulphur dioxide, the oil collects in the pan attached to the float valve (Fig. 147), from which it is constantly withdrawn into the suction line of the compressor.

An interesting type of compressor with eccentric drive is shown in Fig. 148. The condenser consists of a spiral coil placed

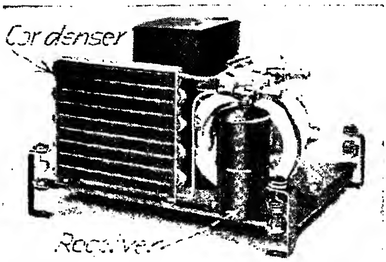


Fig. 146.—Norge refrigerating unit.

around the cylinder of the compressor. An exterior casing

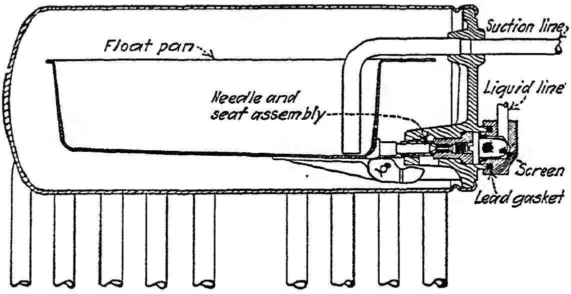


Fig. 147.—Norge float valve.

around the condenser coil provides a space for the circulating

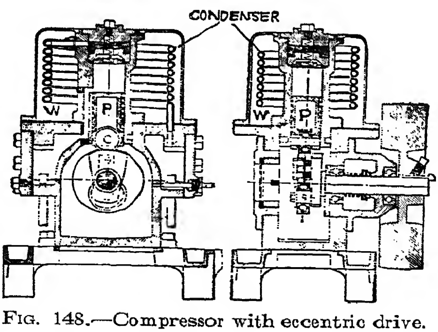


Fig. 148.—Compressor with eccentric drive.

water which is used to cool both the condenser and the cylinder of the compressor.

Icemaster Household Refrigerating System.—The Icemaster household refrigerating unit is shown in Fig. 149. The reciprocating twin compressor is driven by an electric motor by means of a belt. A fan located on the motor shaft provides the air circulation for the condenser which is of the radiator type (p. 154). The refrigerant used is *methyl chloride*. The electric motor used in the domestic sizes is ¼ horsepower, but there are commercial sizes varying from ½ to 2 horsepower. The compressor cylinders of all sizes are air-cooled.

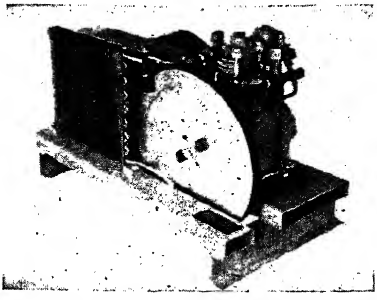


FIG. 149.—Icemaster household refrigerating unit.

In the refrigerating unit shown in Fig. 150, the reciprocating compressor and the electric motor are not direct connected, and the motor drives the compressor *C* by means of reduction gears in the gear box *G*. A fan on the shaft of the motor provides air circulation for the condenser coil *R* of the radiator type.

Removing Air from Small Compression Refrigerating System. When a household unit is installed, it is necessary to remove all air from the various parts of the system. In order to remove it, a by-pass pipe must be provided between the condenser shut-off valve and the suction shut-off valve. The charging drum is connected into the system at some point in this by-pass line. A suitable pipe fitting (usually a tee) must be provided for con-

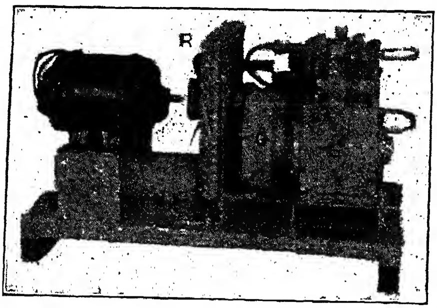


FIG. 150.—Compressor operated with gear box to reduce motor speed.

necting the charging drum into the by-pass line. In order to remove the air in the system by "pumping down" with the compressor, it is necessary to have only a small opening through the main discharge shut-off valve.[1] The shut-off valve at the condenser as well as also the suction shut-off valve may then be opened slightly. The valves are now set so that the air in the condenser, the float-valve chamber, and the suction piping can be drawn through the by-pass line into the compressor and discharged from the system through the small opening in the discharge shut-off valve. The air in the piping which connects the charging drum to the by-pass line will be removed during this process. The air in the cooling coils of the evaporator will also be drawn through the suction valve and will be discharged from the system. If the system includes a cooling unit which is filled with brine, or other solution, not readily frozen, as a part of the evaporator, the filler cap must be removed, so that if there are any leaks in its expansion coil, the cooling unit will not collapse. The compressor may be started and its operation continued until the suction pressure is pumped down to a vacuum of approximately 27 to 28 inches of mercury, when the compressor may be stopped, for a few minutes. If the system holds this vacuum, it is tight.

It is very important that the system should be charged with the proper quantity of refrigerant. This quantity will vary with the size of the system and the distance of the compressor from the cabinet of the refrigerator. After having obtained the required vacuum in the entire system, it is important not to permit any air to enter during the actual charging of the system. To be sure of this, the refrigerant must be put into the system before any valve or connection in the system is disturbed. The charging drum containing the refrigerant having been connected to the suction valve through the tee in the by-pass connection, the valve on the charging drum can be slightly opened to permit the refrigerant to enter the system until the *gage* pressure increases to about 10 to 15 pounds per square inch. When this pressure is reached, the valve on the charging drum may be closed. The discharge shut-off valve may then be opened as well as the condenser shut-off valve. When the valve on the charging drum

[1] Some small compressors have a main discharge shut-off valve with a plug in it which may be removed to obtain a small opening through the valve when it is closed.

is again opened, the pressure of the refrigerant in the charging drum will force the refrigerant through the compressor and into any such device for pressure control as a sylphon bellows. Now, if the main electric switch on the motor is closed, and if the pressure within the system is great enough, the control switch will start the motor and the compressor. When this charging operation starts, the valve on the charging drum should be opened only slightly. As the refrigerant is drawn from the charging drum, it will become chilled, and frost will accumulate, unless the charging drum is placed in a pail of warm water. Care should be taken that the *gage* pressure of the vapor of the refrigerant in the condenser does not exceed 30 pounds per square inch. If it should get above this limit, it may be reduced somewhat by removing the charging drum from the hot water and allowing it to cool. When the system is completely charged, the valve on the charging drum should be closed, and the operation of the compressor continued, to test the automatic starting and stopping. In all this charging operation, it is necessary to watch the gage in order to determine just how the pressure-control switch is working. If the pressure control is satisfactory, the suction shut-off valve may be opened, and the bypass connection may be removed if there is a sufficient amount of refrigerant in the system. The quantity of refrigerant in the system can be ascertained by weighing the drum before and after charging. The whole installation should be thoroughly tested for leaks.

Remote Control of Refrigerating Unit.—When the cooling unit of a household type of refrigerator is located some distance from the compressor and its condenser, the equipment is called a "remote installation," if a suitable control switch is provided near the cooling unit for regulating the refrigerating effect.

As such installations are usually laid out, the compressor, condenser, and electric motor are in a group in the basement or cellar of the building and are entirely separate from the cooling unit. The advantages of this arrangement are, in the first place, that the compressor will operate more economically because of the cooler and more freely circulated air in the basement compared with the conditions that would exist in a confined space that might be provided in the refrigerator cabinet. The temperature of the air in the basement or cellar of a building is usually from 10 to 15° F. cooler than the air in most kitchens and pantries where the refrigerator cabinet is most likely to be located. The

other important advantage is that by locating the compressor equipment in the basement or cellar, there is usually a greater accessibility for inspection and repairs than there would be in a cabinet. The only important disadvantage of this method of installation is that it makes of the refrigerator equipment a permanent installation compared with the portability of a refrigerating equipment in which the compressor, condenser, and electric motor are all located in the cabinet.

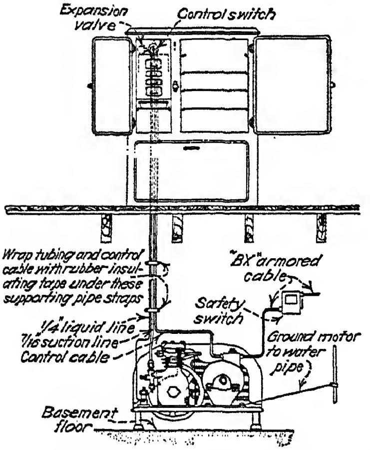


FIG. 151.—Typical remote installation of household equipment.

When the compressor, condenser, and electric motor are installed separate from the refrigerator cabinet in a basement or cellar, precautions should be taken to select a location which is as clean as possible, and the group should be placed so that no water may accidentally be splashed upon them. They should preferably be on a platform sufficiently raised above the floor to avoid the possibility of being flooded if water enters the basement.

A diagram of a typical remote installation for a household refrigerator is shown in Fig. 151. It will be noted that the control switch (p. 174) and expansion valve (p. 173) are marked at the top of the diagram.

Portable Refrigerating Units.—The popularization of mechanical refrigeration created demands for small refrigerating units which do not depend on "outside" sources of energy. For this demand, portable and semi-portable absorption (p. 19) devices have been developed; the device consisting essentially of a "hot ball" and a "cold ball." When in operation to produce refrigeration, the hot ball is exposed to the air and acts as the *absorber* of the refrigerating system, the cold ball being then the *evaporator.*

The history of portable absorption machines goes back to 1865, when Carré brought out a device of this kind in France. Little was done in the commercial development of this type of refrigeration until about 1925, when such a refrigerating device was made

commercially available in that country and, about 2 years later, also in the United States. A typical French machine of this type is shown in Figs. 152 and 153. As illustrated in Fig. 153 it operates on the "intermittent"-absorption cycle in which aqua ammonia is heated in the boiler *B* (generator and absorber), the refrigerant gas passing up around the baffles *b* and through the tube T_1 to the "cooler" *A* (condenser and evaporator), where the condensation of the vapor of the refrigerant takes place and the liquid refrigerant collects.

When the heat applied to the boiler *B* reaches a temperature of 260° F., the indicator *I* drops as the result of the melting of some fusible metal in the tube *t*. At this signal, the application of heat

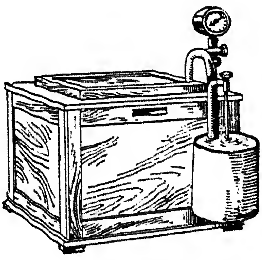


Fig. 152.—Portable absorption refrigerating unit.

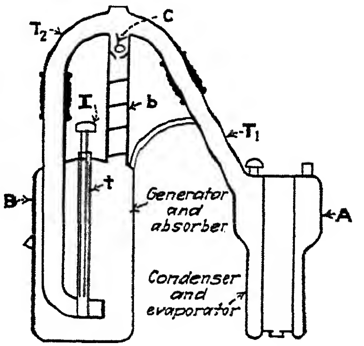


Fig. 153.—Generator and absorber of portable refrigerating unit.

is stopped, and the boiler is submerged in water in a tub or other suitably convenient vessel. This cooling by the water causes the pressure in the boiler to fall, and since the check valve *C* is closed, vapor of the refrigerant from the cooler *A* passes to the boiler *B* (generator) through the pipe T_2, from which it bubbles up into the weak liquor and is absorbed. After a few minutes of cooling in the water bath, the pressure in the boiler *B* falls to a point where refrigerating temperatures are produced in the cooler *A*, and the apparatus is then ready to be placed in a cabinet where it will serve for refrigeration. It can be successfully used in this way for freezing ice.

The heat required for heating the boiler can be obtained by burning gas, kerosene, alcohol, or a similar fuel. Before commencing each heating operation, it is necessary to drain the

condensation from the cooler A into the boiler B by leaving it for 5 or 10 minutes in such a position that the "cooler" is elevated above the level of the boiler.

Icy-ball Refrigerating Device.—An American-built refrigerating apparatus called "icy-ball," operated by a similar method, as illustrated in Figs. 154–158 was designed to produce refrigeration

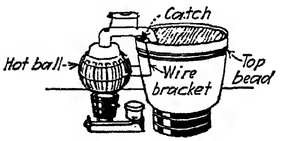


FIG. 154.

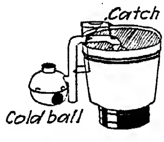


FIG. 155.

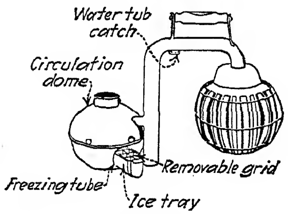


FIG. 156.

FIG. 154.—Icy ball being heated.
FIG. 155.—Icy ball being cooled.
FIG. 156.—Ice tray in icy ball.

more efficiently than the French types that have been explained. It weighs about 36 pounds and has an ice-melting capacity per "heat" of about 16 pounds. It is intended to be heated once a day in average summer weather.

In seeking ways to increase the capacity per unit of volume of similar French machines, the designers adopted the following

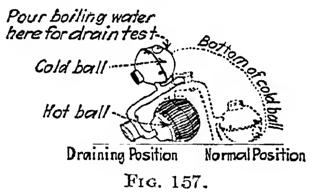


FIG. 157.

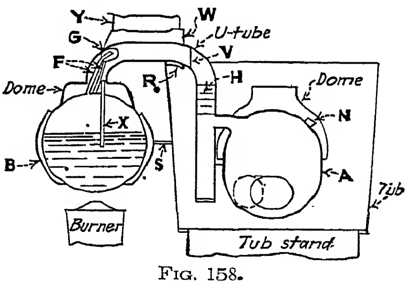


FIG. 158.

FIG. 157.—Draining and normal positions of icy ball.
FIG. 158.—Details of icy ball.

changes: (1) use of spherical shapes to obtain maximum weight of "charge" per unit of weight and volume; (2) increase of the ammonia concentration; (3) heating the boiler (hot ball) to a higher temperature than was customary in the other types. For a given pressure, the concentration could be increased by lowering the temperature of the aqua ammonia during the

absorption period, and this was effectively done by the addition of radiation fins on the side of the boiler (hot ball), as shown in Fig. 158. To obtain an advantage by heating the boiler (hot ball) to a higher temperature, adequate dehydration was provided by the method of adding a dome to the boiler *B* to the top of which the U-tube is attached. In this dome, a "strong" aqua-ammonia solution gathers and is boiled by the hot gases rising from the boiler below. Condensed water vapor and ammonia collects in the lower part of the U-shaped tube and is later returned to the boiler by the method to be explained. A non-return liquid seal is formed by the tubes *F* and the pocket *G*, the latter serving to absorb the surges of liquid refrigerant that occur when the refrigerating unit is handled. When this liquid seal has been formed, the vapor of the refrigerant returns to the boiler *B* through the tube *X*, and when bubbling up through the liquid is rapidly absorbed. The dome on the cooler *A* provides a circulation of water during the condensing period. The brackets *R* and *S* serve for hanging the refrigerating unit on the side of the tub. The boiler *B* and the cooler *A* are each made of two steel hemispheres about 0.08 inch thick and 10 inches in diameter. These units are tested with hydraulic pressure to 600 pounds per square inch. A safety device *N* consisting of the silver disk 0.0025 inch thick opens if excessive pressure develops. A needle valve *W* used in the initial charging of the refrigerating unit, which serves also as a handle bracket, is at the top of the U-tube. The other handle bracket *Y* is flared out so as to form two feet to support the refrigerating unit during the draining operation illustrated in Fig. 157. The charging operation at the factory includes pumping out the air in the system with an air pump so that a relatively high vacuum is established, and then putting the charge of refrigerant into the boiler (hot ball).

The first step in the operation of the "icy-ball" refrigerating device is to place the unit in the draining position (Fig. 157), so that the "cooler" (cold ball) will empty into the boiler. This draining is very important, because if any aqua-ammonia solution is left in the cold ball, it will not only raise the temperature in that part of the unit during the refrigerating period but will retain several times its weight of ammonia in solution at the end of the cycle. The next step is that of heating the hot ball, which is started by attaching the unit to the side of the tube containing the water to be used for condensing the refrigerant vapor in the

cold ball. Just as soon as the cold ball is in the water used for condensing, the burner can be lighted under the hot ball. About 1½ hours is needed for the heating period, overheating or underheating reducing the refrigerating capacity. Indefinite overheating has, however, no permanently troublesome effects on any of the parts, unless the temperature becomes so high that it burns off the galvanizing from the hot ball. In case the heating operation is far too rapid, the safety disk will open. When the heating operation is completed, the burner is removed and the unit lifted away from the tub, so that the hot ball (boiler) can now be lowered into the water. The unit is shown in this position in Fig. 155; the pressure now falls so rapidly that in 10 or 15 minutes the cold ball will reach a freezing temperature.

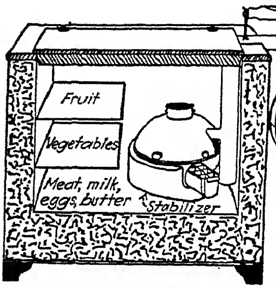


FIG. 159.—Cabinet for icy ball.

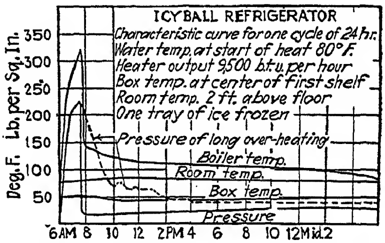


FIG. 160.—Tests of icy-ball equipment.

A cabinet supplied with the icy-ball equipment is shown in Fig. 159. This is a container shaped to fit around the lower half of the cold ball ("cooler"), and contains 15 pounds of anti-freeze solution of the kind commonly used in automobile radiators. This solution acts as a "thermal flywheel," by the method of transferring heat to the cold ball during the early stages of the refrigerating period and then later taking back heat from the air in the cabinet during the latter part of the refrigerating period, and also during the time that the icy-ball equipment is being prepared for another heat.

A typical operating cycle under test conditions of the icy-ball is shown in Fig. 160, showing the operating characteristics of the equipment for one complete cycle of 24 hours. When the outside-air temperature varied from 80 to 90° F. during the cycle, the temperature inside the cabinet varied from 40 to 50° F. and rose to a peak at 55° F. when the icy-ball was being heated. At

the same time, one tray of ice was frozen from water at an initial temperature of 80° F. The pressure in the icy-ball reached a maximum value of 225 pounds per square inch about three-fourths of the way through the heating period and was about 220 pounds per square inch at the end. Calorimeter tests show that the heat capacity of the equipment is about 2,300 B.t.u. per "heat."

This type of equipment is especially intended for places where there is a demand for modern refrigeration, and where electricity or gas services are not available.

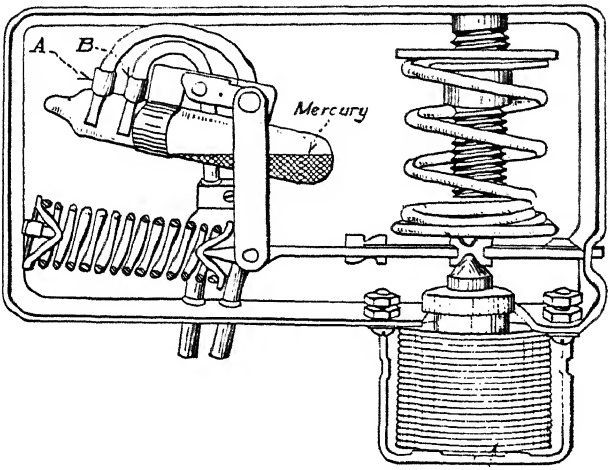


FIG. 161.—Mercoid electric switch.

Thermal Electric Switches.—Some electric switches are operated from a bimetallic thermostat; but are not successful, because it is difficult to make them operate on a temperature *range* as low as 4 or 5° F. An improved switch having a sylphon bellows operated by vapor pressure, called *mercoid control*, is shown in Fig. 161. This type of electric switch has recently found considerable application in refrigerating devices. It consists of a glass tube which contains a small amount of mercury that flows from one end to the other. When the mercury is at the left-hand end of the tube, the electric circuit is completed through the contact points *A* and *B*, and when it is tilted so that the mercury is at the right-hand end, as shown in the figure,

there is no connection between A and B, and the circuit is broken. In this way, a quick make-and-break contact is secured. In order to avoid the corrosion produced by the arcing of the spark when the circuit is broken, the glass tube containing the mercury is filled with an inert gas in which an electric spark produces no combustion or oxidation.

Electrolux Servel Gas Refrigerator.—The gas-heated refrigerator made by the Electrolux Refrigerator Sales, Inc., is an example of the absorption-system type of refrigeration as applied

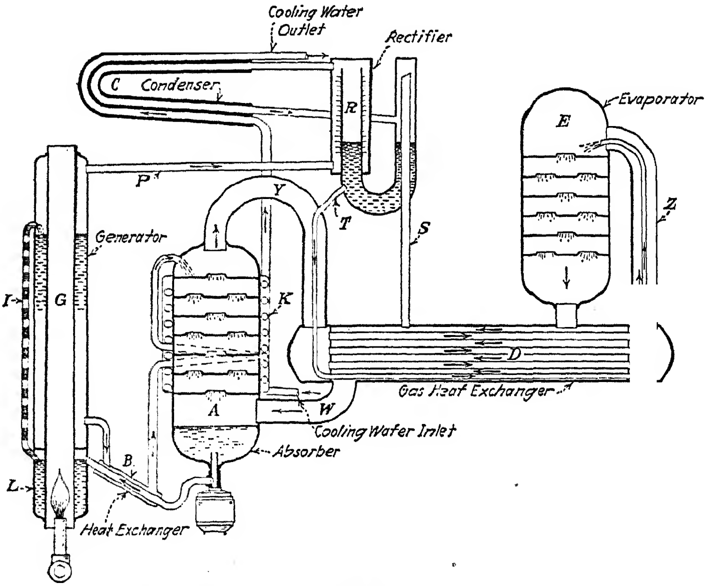


Fig. 162.—Servel gas-heated refrigerating unit.

to household purposes. For many years, the gas engineer has been looking for a household refrigerating system that would utilize gas instead of electricity for its energy supply. This kind of refrigerating unit is highly desirable for the gas manufacturer, as the demand for gas for such refrigerators balances seasonably, to some extent, the requirements for house heating in the winter months. The refrigerant is ammonia, which is not considered so safe for household use as some other refrigerants.

The refrigerating unit, as shown in Fig. 162, consists of a generator G, rectifier R, condenser C, absorber A, liquor-heat

exchanger *B*, gas-heat exchanger *D*, and evaporator *E*. Heat is supplied to the generator *G* by the gas flame of a Bunsen burner, which heats the strong ammonia liquor at the bottom of the generator. This heat causes an increase in pressure in the lower section *L* of the generator, which forces slugs of strong ammonia liquor to rise through the interconnecting line *I*, thus feeding strong ammonia liquor into the top of the generator. When still more heat is added to the strong liquor in the generator, a mixture of ammonia and water vapors passes from the generator through the pipe *P* into the rectifier *R*.

When the strong liquor in the generator has been reduced in concentration, a weak liquor is thus formed. The weak liquor, because of the difference in level, is forced through the liquor-heat exchanger *B* into the top of the absorber *A*.

The ammonia and water vapors formed in the generator enter the rectifier *R* in which heat is removed from the mechanical mixture of the vapors by the evaporation of liquid ammonia, thus causing the water vapor to condense. The water thus formed by condensation then absorbs ammonia vapor, so that an ammonia solution of strong concentration is returned to the generator *G*, through the pipe *P*.

The rectifier *R* is used, therefore, to separate the water vapor from the ammonia vapor, permitting the ammonia vapor to pass through to the condenser *C* where the cooling water removes a sufficient amount of heat to cause the ammonia vapor to liquefy. In the liquid state, the ammonia flows from the rectifier *R* through a pipe *T*, which passes through the lower part of the gas-heat exchanger *D* and discharges the liquid ammonia into the top of the evaporator *E*. In the evaporator, the liquid ammonia flows over trays which have openings permitting it to fall over one row of trays after the other until it is evaporated. Hydrogen gas enters the top of the evaporator *E* through *Z* and mixes with ammonia vapor, causing the partial pressure of the latter within the evaporator to be low enough to obtain a low temperature. The law (Dalton's)[1] of *partial pressures* is thus utilized, as the total pressure in the evaporator is the sum of the pressures of the hydrogen gas and the ammonia vapor. By admitting the hydrogen gas along with the ammonia to the evaporator, the partial pressure of the latter will be lower than if the ammonia vapor

[1] For an explanation of Dalton's law, see "Elements of Engineering Thermodynamics" by Moyer, Calderwood and Potter, 5th Ed., pp. 22–25.

was used alone, thus giving a low evaporating temperature. After the liquid ammonia has evaporated, the ammonia vapor and the hydrogen gas, acting as a mechanical mixture, pass downward through the gas-heat exchanger *D*, where they cool the liquid ammonia flowing from the rectifier to the evaporator through the pipe *T* as well as the hydrogen gas which is passing into the evaporator.

The mechanical mixture of hydrogen gas and ammonia vapor, after passing through the gas-heat exchanger *D*, enters the bottom of the absorber *A* through the passage *W*. The ammonia then rises and mixes with the cool weak liquor entering the absorber at the top. This weak ammonia liquor flows over trays in the absorber which have numerous small holes, so that the weak liquor falls like rain and mingles with the rising mechanical mixture of ammonia vapor and hydrogen gas. The concentration of the weak liquor is such that it readily absorbs ammonia vapor, thereby causing the strength of the solution to increase so that it becomes a strong liquor. The strong liquor accumulates at the bottom of the absorber, and because of a difference in pressure, its vapor flows through the passage *Y* into the left-hand end of the gas-heat exchanger *D*. In the heat exchanger *B*, heat is absorbed from the weak liquor, which is on its way to the absorber, thus increasing the temperature of the strong liquor about to enter the generator *G*.

Cooling coils *K* in which water is circulated are provided to remove the partial heat of absorption generated by the absorption of ammonia vapor by the weak liquor. These cooling coils are made of copper and are placed around the exterior of the casing of the absorber *A*. The hydrogen now left free of ammonia is allowed to pass through the tubes of the gas-heat exchanger *D* and the passage *Z* into the evaporator *E*.

The condenser *C* consists of adjacent copper and steel coils, the copper one containing water, and the other ammonia. These coils are in direct contact, and heat flows by conduction from the ammonia to the water. In order to obtain large contact surfaces, a copper wire is soldered to the adjacent coils in order to increase the rate of heat transfer. The cooling water is supplied at the bottom of the cooling coils *K* of the absorber *A* and is discharged directly into the cooling coils of the condenser. The temperature of the cooling water at the outlet of the condenser is maintained at about 90° F. by permitting a limited quantity of water to

pass through the cooling coils. This temperature is controlled entirely by a thermostatic water control. The quantity of water required is about 2 to 8 gallons per hour. This quantity depends on the difference between the inlet and the outlet temperatures of the water; and also on the "room" temperature.

The generator, evaporator, absorber, rectifier, and gas-heat exchanger are made of heavy steel tubing interconnected by steel pipes. All the joints are made by oxyacetylene welding. These parts will withstand a pressure as high as 3,000 pounds per square inch, although the maximum charging pressure is only

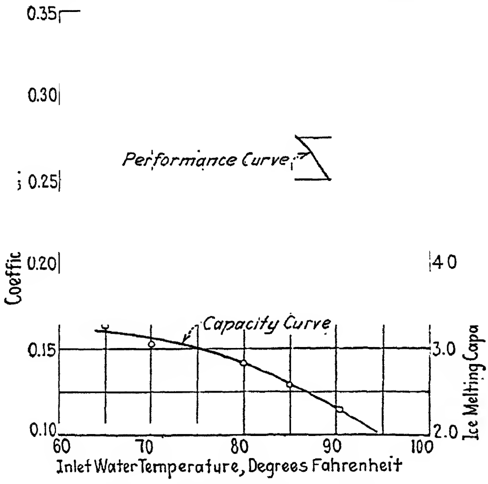


FIG. 163.—Effect of varying cooling-water temperature. Gas rate 2.5 cubic feet per hour. Room temperature 70° F. Quantity of water 7 gallons per hour.

about 200 pounds per square inch.[1] An apparatus of this kind requires an automatic mechanism which is regulated by temperature for shutting off the gas supply at the end of the boiling period and, also, for adjusting at the right time the cooling water going to the condenser and to the absorber.

This system has a very satisfactory safety feature, in that the radiating surface (the outside surfaces of the evaporator, heat exchangers, absorber, condenser, and rectifier) is large in proportion to the heating surface. These will dissipate heat by radiation almost at the same rate that it is supplied to the generator.

[1] In order to prevent any destructive effect of the ammonia on the metal parts, a little ammonium bichromate, $(NH_4)_2Cr_2O_7$, is added to the aqua ammonia which circulates in the system.

A fusible plug is fitted into the bottom of the absorber, in order to provide an emergency outlet for the ammonia in case of exterior heating by the outbreak of fire in the room or building. This fusible plug is made of a metal which melts at 200° F.

Figure 163 shows the effect of the coefficient of performance and ice-melting capacity for varying inlet-cooling-water temperatures when the rate of gas consumption is 2½ cubic feet per hour, the room temperature is 70° F., and 7 gallons of water are used per hour. It is interesting to note that the curve of coeffi-

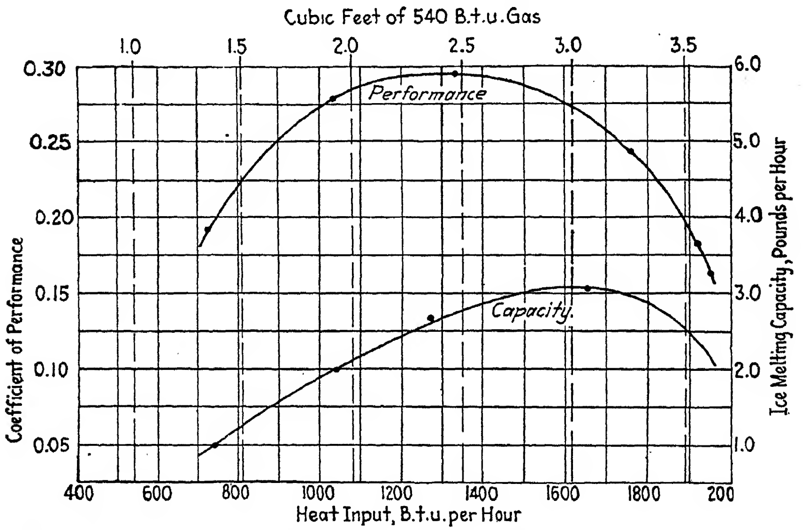


FIG. 164.—Effect of varying amount of heat supplied. Water temperature 70° F. Room temperature 70° F. Quantity of water 5 gallon per hour.

cient of performance falls off rapidly at temperatures of inlet cooling water above 80° F.

Curves showing how the coefficient of performance and the ice-melting capacity are affected by the amount of heat supplied are given in Fig. 164. These curves are based on a heating value of the gas of 540 B.t.u. per cubic foot. The coefficient of performance has its maximum value when the total heat supplied is about 1,300 B.t.u. per hour, while the ice-melting capacity has its highest value when the total heat supplied is about 1,620 B.t.u. per hour. The rate of gas consumption for average operation is shown by the curves to be between 2 and 3 cubic feet per hour.

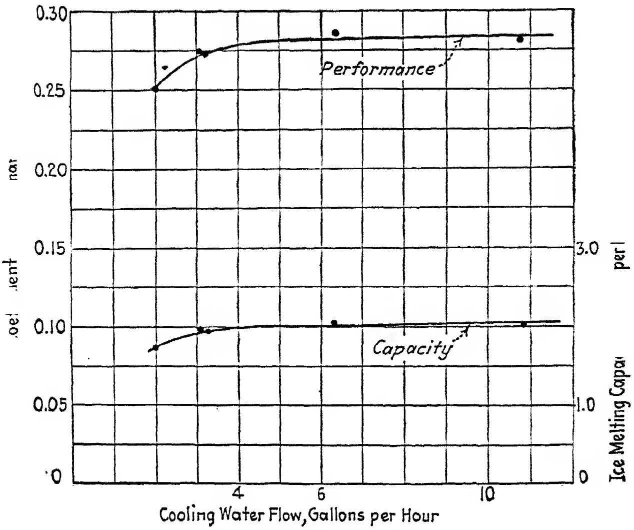


FIG. 165.—Effect of varying quantity of cooling water. Cooling-water temperature 70° F. Room temperature 70° F. Heat input 1,025 B.t.u. per hour.

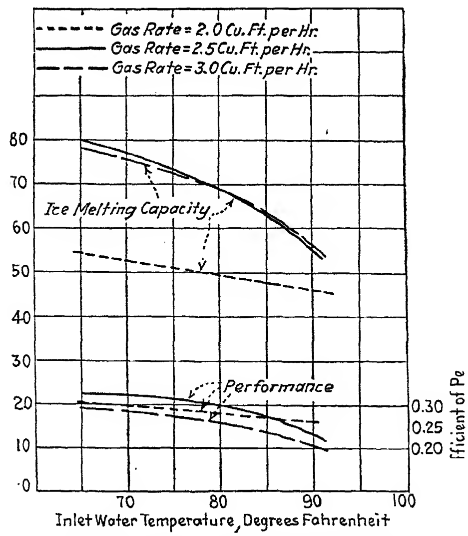


FIG. 166.—Effect of varying cooling-water temperature for three rates of gas consumption. Room temperature 70° F. Quantity of water 7 gallons per hour.

A curve showing how the coefficient of performance and the ice-melting capacity are affected by the quantity of cooling water in gallons is given in Fig. 165. For rates of flow of cooling water of about 4 gallons per hour and above, the coefficient of performance and ice-melting capacity do not change appreciably.

In Fig. 166, there are curves showing the effect of cooling-water temperature upon efficiency and ice-melting capacity of the refrigerating system, for three rates of gas consumption. It is interesting to note the wide divergence in the values of ice-melting capacity for gas rates of 2 and 3 cubic feet per hour.

Some of the important factors in the operation of this refrigerating system are: (1) The maximum performance is obtained when the heat input is about 1,300 to 1,350 B.t.u. per hour, which is approximately equivalent to 2.4 cubic feet of gas per hour; (2) the lower the average temperature of the cooling water the better is the performance; (3) refrigeration for domestic purposes may be obtained when the temperature of the cooling water is 90° F. and the room temperature is 100° F.; (4) maximum capacity of the machine is obtained when the heat input is about 1,620 B.t.u. per hour, which is equivalent to approximately 2.9 cubic feet of gas per hour;[1] (5) room temperature affects the efficiency of the machine slightly but not enough to interfere with its operation for the ordinary range of temperature; (6) in case the cooling water fails and the gas continues to burn, refrigeration will stop, and the maximum pressure will not increase more than about 25 per cent above the normal working pressure.

A domestic refrigerator cabinet with a volume of 5 cubic feet, maintaining in its cooling coil a temperature of 17° F. when the "room" temperature is 80° F., consumes about 2.6 cubic feet of gas per hour, the gas having a heat value of 540 B.t.u. per cubic foot. The quantity of water per hour varies with the difference in temperature between the inlet and the outlet of the cooling water. For the following differences in temperature of the cooling water, in degrees Fahrenheit 15, 20, 25, and 30, the gallons of water per hour are, respectively, 6.8, 5.0, 4.0, and 3.0. In the case of a smaller cabinet, a smaller amount of cooling water is required.

[1] The capacity for this heat input is approximately 3.3 pounds of ice-melting capacity per hour, which is equal to about 79 pounds of ice-melting capacity per 24 hours. This represents a daily gas consumption of about 70 cubic feet.

Gas Thermostat.—The gas-control device used in the Electrolux refrigerator is shown in Fig. 167. In essential parts, it consists of a thermal bulb *A* which controls the quantity of gas required to maintain the desired refrigerator temperatures, and a thermal bulb *B* which operates to close the gas valve when the water supply fails. These thermal bulbs connect to two "diaphragm" chambers. The upper one shown in Fig. 167 is connected to the bulb *B*, while the lower is connected to the bulb *A*. As the pressure in the bulb *A* rises with an increase of temperature in

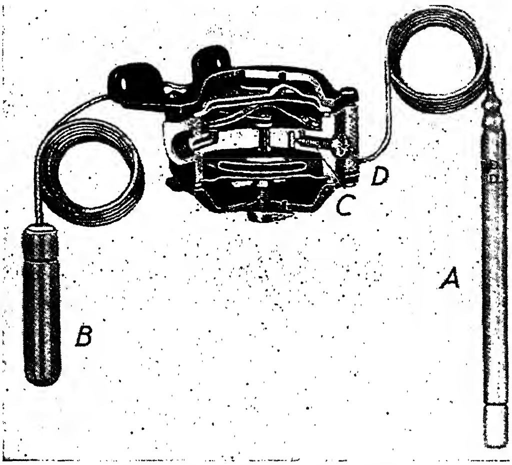


Fig. 167.—Gas-control device.

the refrigerator cabinet, the pressure inside of the lower diaphragm chamber increases and by bulging opens the gas valve. Since the gas control must supply some gas continuously to the burner, a by-pass valve *C* permits enough gas to enter the burner (Fig. 168) to keep it lighted. This in no way affects its operation, except to supply enough heat to the system to take care of the heat losses when no refrigeration is needed. Since the gas-pressure range of the gas thermostat is limited, it is often necessary to connect into the gas line (ahead of the refrigerator) a gas-pressure-reducing valve.

Burner.—The burner is of the Bunsen type having been modified to suit the needs of this application. The supply of gas admitted to the burner is regulated by an adjusting spring shown

in Fig. 168. This spring regulates the position of the needle valve with respect to its seat. In case the burner flame should happen to be extinguished the gas supply must be automatically shut off to prevent the escape of gas into the room. To accomplish this a thermostat, made of two metals back to back and having suitable coefficients of expansion to cause a change in shape on heating, is used to close a small supply valve located in the burner. This thermostat is circular in shape, but it is also slightly dished. When heated, the thermostat changes its curvature from concave to convex and on cooling returns to its initial shape. This thermostat is located in a little chamber

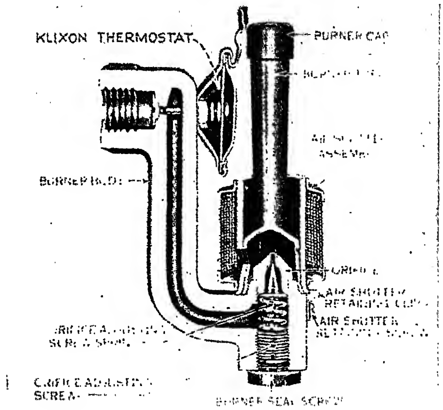


FIG. 168.—Gas burner for refrigerator.

shown in Fig. 168 to which is connected a heat-conducting element, rising to the upper part of the burner. This element conducts heat from the burner down to the thermostat causing the valve to open, thus allowing the gas to enter the burner. If the burner flame is extinguished, the thermostat cools and closes the valve. This operation takes place in about 30 seconds. In order to again start the flow of gas, it is necessary to apply heat to the burner and heating element. This may be done by the use of a little pilot light with a suitable valve for admitting more gas to permit the pilot flame to come in contact with the burner.

Thermostatic Water Control.—Since it is desirable to make the Electrolux refrigerator economical in operation the water supply

should be controlled. This is accomplished by the use of the water-control valve shown in Fig. 169. This valve and closed by the temperature of the discharged cooling water. Heat is conducted from the discharged cooling water to the sylphon bellows by conduction through the nut located in a rubber diaphragm which separates the water from the bellows. The pressure in the bellows changes with the change in temperature of the discharged cooling water, causing the bellows to expand or contract. The temperature of the discharged cooling water is generally held at about 90° F.

Water Lines.—In private homes it is customary to connect the refrigerator water line to the water main in the basement. In this line a suitable strainer and pressure regulator are located. The strainer serves to prevent any scale from entering the refrigerator system, and the pressure regulator reduces the water pressure to the desired pressure for satisfactory operation of the refrigerator. The pipe carrying the discharged condenser water is generally connected directly to the sewer and has a check valve located just inside the sewer connection.

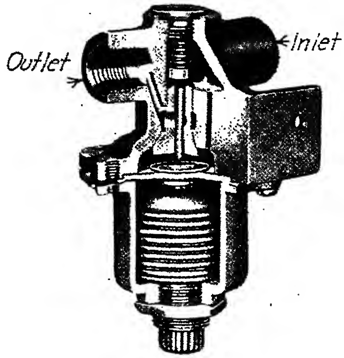


FIG. 169.—Water-control valve.

Temperature Control.—It is not customary to use a hand-operated temperature-control device with Electrolux refrigerators. However, in some large cabinets, manually controlled temperature controls are provided.

Cabinets.—The cabinets are lined with white vitreous porcelain fused on rust-resisting metal, while the steel exterior is finished with white lacquer. For insulating material corkboard is used, treated with hydrolene to keep out moisture. The insulation varies in thickness from 2 to 3 inches. Figure 170 shows an Electrolux refrigerator cabinet and gas range combined for the requirements of small apartments.

Faraday Gas Refrigerator.—Recently the Faraday Refrigerator Company (General Motors Corporation) has placed on the market a gas refrigerator that resembles somewhat the type of

refrigerating apparatus developed by Michael Faraday in 1823. Faraday used ammonia for the refrigerant and silver chloride for the absorption material as silver chloride will absorb large amounts of ammonia vapor. When powdered silver chloride has absorbed all the ammonia gas that it will hold and when the material is heated, the ammonia gas is driven off. If the equipment is so arranged that this hot gas is under pressure, then, when its heat is removed, the ammonia vapor changes into the liquid state. Now if the source of heat is removed and the silver chloride is allowed to cool, the latter is capable of again absorbing ammonia vapor. The chamber that is filled with the liquid ammonia acts as a refrigerating unit because the liquid ammonia as it evaporates takes up heat, thus cooling this chamber and the air that comes in contact with it. This process may be repeated at intervals as refrigeration is required.

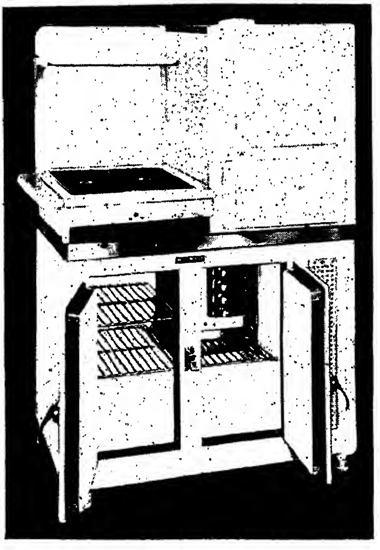


FIG. 170.—Combined gas-heated frigerator and gas range.

The Faraday refrigerator operates like the above apparatus. It consists of an absorber, condenser, chilling unit, and controlling devices. The absorbing agent is strontium chloride, and ammonia is used as the refrigerant.

The absorber is made of a series of steel disks electrically welded together and is completely sealed. It has steel partitions which extend into the absorbing material of the vapor jacket. All parts of the absorber exposed to the flame of the gas burner are treated with lead to prevent corrosion. The entire absorber is completely enclosed in an aluminum shield. This shield, because of the peculiar properties of aluminum in reflecting heat, greatly reduces heat losses in the absorber, due to radiation.

The water-cooled condenser is composed of coils of copper tubing, formed in such a manner that maximum advantage is taken of the cooling surfaces. The separate tubes are soldered

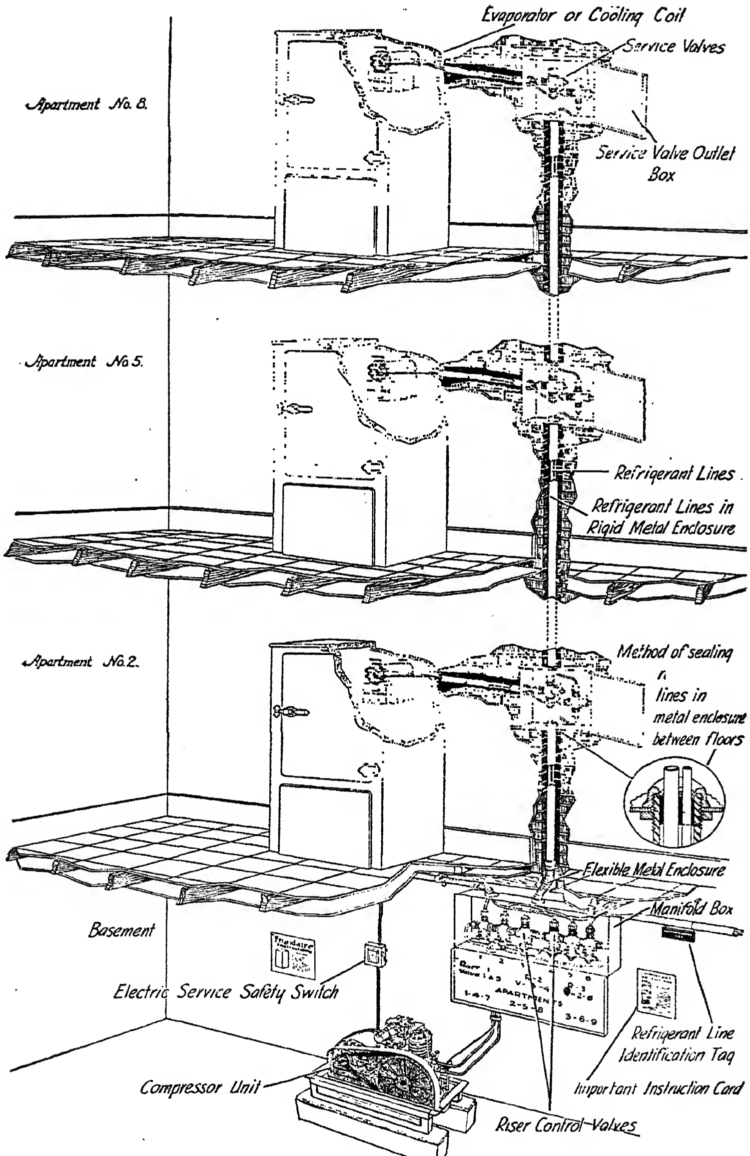


Fig. 171.—Multiple-unit household installation.

together to promote a rapid interchange of heat. A thermostatic water valve is located in the cold-water inlet.

The gas burner is provided with a gas filter, a pressure regulator that delivers gas to the burner at a constant pressure, and a thermostatic pilot of the Bunsen type that shuts off the entire gas supply if for any reason the pilot flame is extinguished. An anti-flashback burner is used which is made of stainless steel to prevent corrosion.

Under normal operating conditions in a room at 70° F., and with the cooling water at 60° F., about 1,300 to 1,800 cubic feet of manufactured gas, and approximately 300 to 400 cubic feet of water per month are required. These quantities depend somewhat, however, on the size of the refrigerator equipment.

Multiple Household Refrigeration Installations.—In apartment houses, hotels, and clubs, many installations are made by using one compressor with several cooling units. In this type of installation, the compressor is located at some convenient place in the basement. The multiple installation has been generally adopted because of the following reasons: (1) Lower first cost; (2) lower power consumption; (3) lower maintenance cost; (4) lower depreciation; (5) higher investment return. In some states this type of installation is by law considered as part of the building and once installed cannot be removed without court action. Because of this situation manufacturers prefer to install, in certain cases, self-contained or domestic refrigerators.

The number of coils or cooling units which can be used with a given size of compressor will depend upon the size and rating of the coils, temperatures desired, insulation of the cabinets, and the distance from the compressor to the cooling units. A typical multiple installation is shown in Fig. 171.

During the past, many different multiple installations have been made. Each territory has had its own method of making these installations, resulting in both good and bad installations. In buildings under construction different specifications for installing equipment may be followed than in buildings which have been completed. In general, the riser type of installation is recommended; that is, a riser or risers extend from a valve board located near the compressor in the basement to the top of the building, provided the height of the building does not exceed the maximum height allowable for the liquid and suction lines of the compressor. The liquid- and suction-line risers must

each have a valve at the bottom of the riser located on a valve board near the compressor. Valves must also be located in suitable junction boxes and placed in these lines between the riser and the cooling unit at each service outlet, as is shown in Fig. 172. The left-hand riser shows a method of using flexible conduit; the middle, rigid conduit; and the right-hand, combination of flexible and rigid conduit. Some multiple installations have individual liquid and suction lines for each cooling unit. These lines should be fully protected by conduit and have valves placed at the start of each riser as well as at each cabinet.

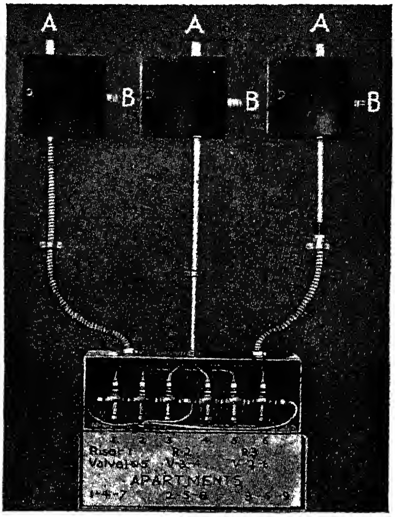


FIG. 172.—Valves in junction boxes.

Multiple installations are generally operated with the flooded system (p. 161); but installations of this type have been successfully operated with the dry system using suitable expansion valves and electric controls.

Household Absorption Refrigerating System Using Water as Refrigerant.—A French household absorption machine invented by R. Follain is interesting because water is the only medium of refrigeration employed. This is doubly advantageous, since water everywhere is cheap and absorbs a larger quantity of heat on evaporation than any other substance known. In this apparatus, the evaporation is hastened, and, therefore, the cooling effect is intensified through the creation of a vacuum above the surface of the water in an airtight tank by the injection of a steam jet in a constricted tube. The water vapor and steam are condensed in an adjoining chamber by a spray of cold water. Several such systems can be arranged in series in order to secure the desired reduction of temperature. Such a machine will cool 1,100 pounds of water from 77 to 37° F. with the use of about 7.9 pounds of water utilized as steam for the injector and 450 pounds of water for cooling at average yearly temperatures.

Importance of Adequate Servicing Arrangements.—Most of the household refrigerators which have been in quantity production for five or more years for national distribution during that time will be reasonably satisfactory in operation, so that the important factors influencing a selection are: (1) Adequateness of servicing facilities, and (2) effect on health of leakages of the refrigerant (p. 73). Most of all, the decision of the purchaser is influenced by the local facilities of the manufacturer for servicing minor repairs and adjustments; the adequacy of servicing depending, of course, on the probabilities of its permanency and its availability without embarrassing and costly delays. When a selection is to be made, the purchaser should have the opportunity to consult a number of persons in his community who have had long experience with the type he is considering for purchase.

CHAPTER VI

OPERATION OF REFRIGERATION SYSTEMS

Operating Practice for Compression System. Starting a Compressor.—The reciprocating engine-driven compressor of a refrigerating system should always be started slowly and carefully, for the reason that the machine is handling powerful chemicals at high pressures. When starting a water-cooled compressor, it is necessary to provide, first, a flow of cooling water through the condenser and also through the water jacket of the compressor. It is during the starting operation that the cylinder of the compressor is most likely to become too hot.

Before steam is admitted to an engine driving a compressor, the main discharge valve at the compressor must be open, while the main suction valve and the valve on the main liquid line *A*, in Fig. 173, must be closed. The packing in the stuffing box of the compressor should be examined to make sure that it is not too tight and that the oil cups supply sufficient oil to the bearings. In the operation of starting, the engine cylinder should be "warmed." If a slide-valve engine is being used, this can be done by opening the drain cocks and letting into the cylinder just enough steam to warm it. The drain cocks should not be closed until the engine has run for a little while. In the case of a Corliss engine, which does not have drain cocks, warming is accomplished by opening the throttle valve of the engine a little and unhooking the wristplate and rocking it back and forth so that a small amount of steam enters each end of the cylinder, being careful that the valves are hooked to the wristplate. Then the wristplate should be attached to the rocker arm and to both dashpot rods. The throttle valve may then be opened only wide enough to bring the engine past dead center; and after operating for a few strokes, the suction valve of the compressor may be opened slightly. If the compressor cylinder "knocks," the suction valve should be closed a little, a knocking sound indicating that liquid refrigerant is entering the cylinder. Since the liquid is not compressible, its presence may cause a cylinder

head of the rigid type to be broken off. If there is no knocking, the suction valve should be opened more and more at intervals until it is wide open. Usually, an increase in the opening of the

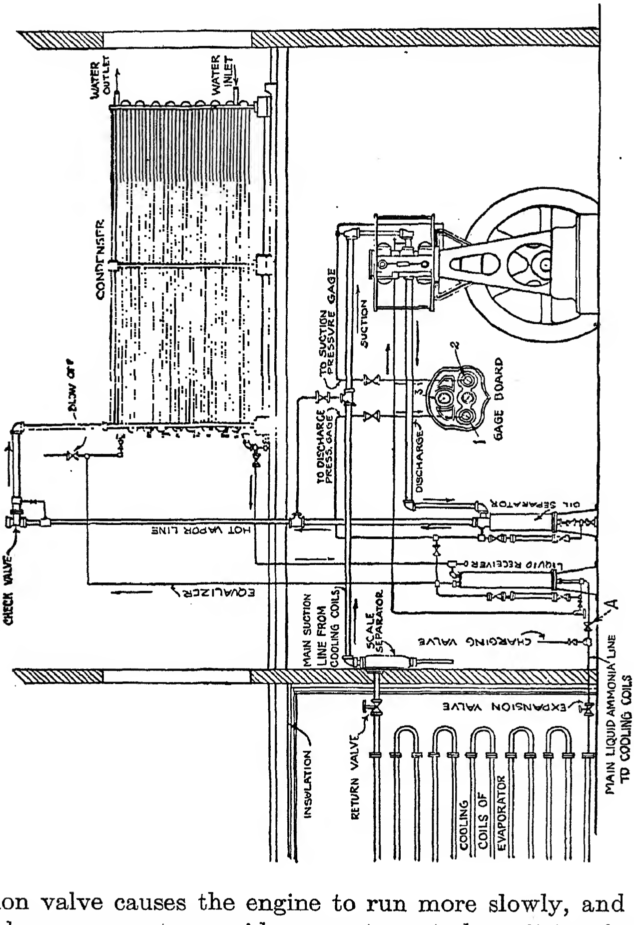


FIG. 173.—Complete ammonia refrigerating system.

suction valve causes the engine to run more slowly, and it will then be necessary to provide more steam to keep it turning over.

Most compressors, and especially those driven by electric motors, are equipped with a by-pass connection so that only a small amount of power will be needed for starting. When starting a motor-driven compressor, the valves on the main discharge and suction lines should be closed, and the valve on the by-pass

connection opened. When the motor is running at low speed, the valves on the main discharge and suction lines should be opened and the valve on the by-pass closed. The speed of the compressor should then be adjusted so as to bring the suction pressure to the desired value, after which the expansion valve can be opened and adjusted to the proper conditions.

In order to shut down a motor-driven compressor, reverse the order of operations as above. When the compressor is motor driven it should be remembered that the power required to operate with high suction pressure is much greater than with low suction pressure. It often happens that the suction pressure is high at the start and because of this about 40 per cent more power is required than under operating conditions.

When a refrigerating plant has been shut down for several hours, the evaporation of liquid refrigerant in the cooling coils of the evaporator will probably generate a high pressure that will be indicated on the suction gage. This pressure will be reduced when the compressor is started and the suction valve is opened. In a plant using ammonia as the refrigerant, for example, until the suction gage registers a pressure of at least 5 or 10 pounds per square inch, the compressor must not be run above three-quarter speed; half-speed is even better. In the meanwhile, the temperature of the discharge pipe of the compressor should be observed. If it gets too hot to be touched comfortably by the hand (110 to 120° F.), the speed of the compressor should be reduced by slowing the engine.

With the expansion valve still closed, the valve on the main liquid line *A*, in Fig. 173, may now be opened, and then the expansion valve should also be opened a little at a time to admit a fine stream of liquid ammonia into the cooling coils of the evaporator.

When the suction pressure increases and the suction pipe becomes cold, the engine may be operated at full speed. It is a good plan to put one's hand on the suction pipe from time to time to observe whether or not it is getting colder, as it should be when the compressor is operating properly.

As the expansion valve is gradually opened wider, more frost will appear on the cooling coils of the evaporator as far back on the suction pipe as the suction valves. This is an indication that the expansion of ammonia into the coils is going on properly. In an ammonia refrigerating system, the expansion valve is

adjusted properly when the suction gage reaches the desired operating pressure.

Temperatures, Pressures in Compression System of Refrigeration.—The vapor of a refrigerant will not be condensed unless its temperature is higher than that of the condenser cooling water. The discharge pressure of the compressor, therefore, must always be high enough to keep the vapor of the refrigerant at a sufficiently high temperature. It is important, however, that there should not be too much difference in temperature, for efficiency of operation requires the smallest possible temperature range. The best effects are obtained when the condenser cooling water is as cold as possible, in order that the discharge pressure of the compressor may be so low that the ammonia will not be at an excessively high liquefaction temperature.

In the cooling coils of the evaporator, the temperature of the vapor of the refrigerant depends on the suction pressure. By adjustment of the suction pressure, then, the temperature of the refrigerant in the cooling coils of the evaporator can be controlled. The suction pressure can be *increased* by two methods: (1) by opening the expansion valve a little or (2) reducing the speed of the compressor. This pressure is *decreased* similarly by closing the expansion valve slightly or increasing the compressor speed. The speed of the compressor and the amount of opening of the expansion-valve control, also, the discharge pressure; but the effects of compressor speed and expansion-valve opening on the discharge pressure are opposite to their effects on the suction pressure. A high discharge pressure is maintained by high compressor speed; and low discharge pressure is maintained by low speed.

If a temperature of 32° F. is to be maintained by a brine cooling system, it will be found necessary to keep the suction-gage pressure in a plant using ammonia as the refrigerant at 25 to 28 pounds per square inch, in order that the ammonia may be enough colder than the brine to insure a rapid transfer of heat from the brine to the ammonia. This difference in temperature should be 10 to 15° F. Similar conditions in a direct-expansion system require a suction-gage pressure of the ammonia vapor of 33 to 35 pounds per square inch.

If it is desired to secure a rapid freezing temperature (0° F. or lower), the suction-gage pressure of ammonia vapor will have to be kept as low as 5 pounds per square inch; while 20- to

25-pounds-per-square-inch gage will give the proper temperature (10 to 20° F.) for making ice.

Shutting Down an Ammonia Compressor.—Methods of shutting down the compressor in an ammonia refrigerating system vary according to the length of time it is desired to keep the compressor out of operation.

To shut down for 2 or 3 hours, the suction-gage pressure should be reduced to not more than 5 pounds per square inch, by operating, if possible, the compressor with the valve on the main ammonia liquid line *closed.* It may happen that the compressor will tend to overheat while thus reducing the suction pressure. In this case, the speed of the compressor should be reduced until the compressor is no longer getting hotter. When the suction gage indicates the proper reduction in pressure, the compressor should be stopped. Then, and not until the compressor has come to a *dead stop,* the main suction and the discharge valves should be closed. It is important that the main discharge valve should never be closed while the compressor is still operating.

After the engine driving the compressor has been stopped, the ammonia vapor will be likely to escape at the compressor stuffing box, unless precautions are taken. To avoid this, it is a good practice to tighten the packing of the stuffing box when shutting down. Finally, the cooling water should be shut off.

For a longer period of shutdown (a day or more), it will also be necessary to close the expansion valve after "pumping down" the suction pressure.

If the shutdown is to last not more than about 1/4 hour, only the valve on the main liquid line *A*, in Fig. 173, need be closed; and when this valve is closed, the engine driving the compressor may be stopped.

Opening the Cylinder of a Compressor.—As in the case of any similar machine, the cylinder of a compressor needs such occasional care as cleaning or regrinding of valves. To remove the valves for this purpose, the main suction valve (marked *S*, in Fig. 174) is closed, and the compressor is operated for about a dozen revolutions. After stopping the compressor, the main discharge valve should be closed, and the blank flange should be removed from the valve *Z*. A pipe connection, as shown at *Y*, should be made from the valve *Z* into a bucket of water. When this valve *Z* is opened a little, the vapor of the refrigerant in the

compressor will pass into the bucket of water, making it possible to open the cylinder.

Oil for Compressors.—The compressor cylinders of a refrigerating plant are subject to unusual conditions of service and need special grades of oil. The oil which is used should be free from animal or vegetable matter and should not freeze, thicken, or gum at low temperatures. High-grade paraffin petroleum oils will meet these requirements satisfactorily in plants using ammonia as the refrigerant.

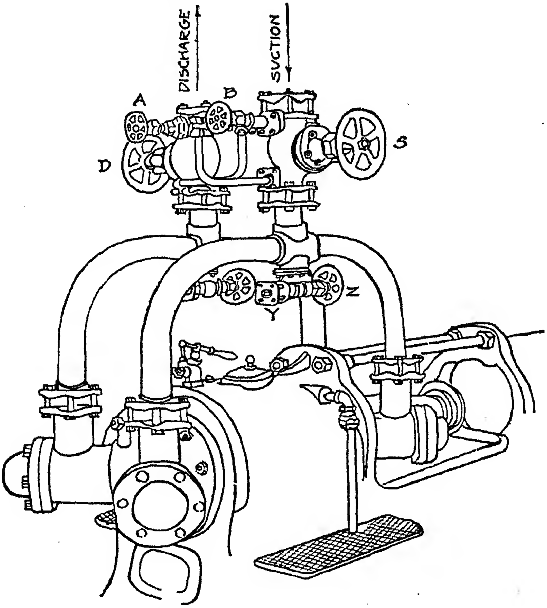


FIG. 174.—Bypass connections on ammonia compressor.

When oil is used too freely in the compressor, it is likely to find its way into the condenser and cause trouble there. It should be used sparingly but in sufficient amount to produce the necessary lubricating effect. Once a week or oftener, the oil which has passed through the oil separator into the condenser or other parts of the system should be cleaned out by the method of circulating hot ammonia vapor through the pipes to make the oil more fluid than it would otherwise be. Then the mixed oil and ammonia should be "pumped" through the compressor into the oil separator.

Oil Separator.—The discharge-pipe line of the compressor shown in Fig. 173 is provided with an *oil separator*. This separator is necessary to remove from the refrigerant the oil supplied to the cylinder of the compressor to lubricate the piston and to remove, also, the small amount of oil provided for the stuffing box and which may leak into the cylinder. Unless removed, some of this oil will remain mixed with the vapor of the refrigerant and will be carried into the condenser where it will collect on the walls of the pipes, preventing the efficient transfer of heat and reducing the effectiveness of the condenser. Briefly, an *oil separator* is used to remove the oil before the vapor of the refrigerant enters the condenser.

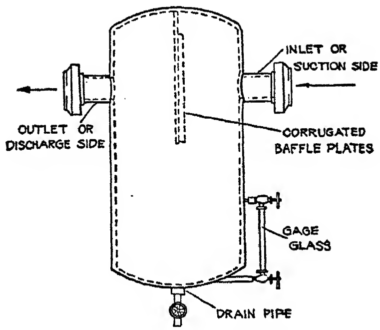


FIG. 175.—Oil separator with baffle plates.

The oil separator, shown in some detail in Fig. 175, is simply a vertical cylinder connected to the discharge pipe of the compressor so that the flow of vapor of the refrigerant is intercepted by baffle plates, as shown. Some of the vapor passes through holes in the corrugated baffle plates, and some vapor passes around the plates. The pipe connections leading, respectively, to the compressor and to the condenser are at the top of the cylinder. The vapor of the refrigerant enters the oil separator at a high velocity, and the sudden change in direction of flow of the vapor, caused by the baffle plates, tends to leave the oil on the baffle plates and the inside wall of the separator, to which it clings. It then trickles down to the bottom, where it collects and is drained off. Some oil separators are made with the discharge pipe of the compressor passing halfway down on the inside, and others have a special passage with fins placed along the path of the vapor for the purpose of separating the oil. An oil separator generally has a gage glass connected to it at one side, as shown in Fig. 175. At the top and bottom of the gage glass are cocks, which, when open, show the depth of oil in the separator. Except when testing the depth of oil in the separator, these cocks should be kept closed to prevent the possible escape

of the refrigerant.[1] From the top of the oil separator, a small pipe usually extends to the gage board, where a gage indicates the pressure in the discharge line of the compressor. Figure 176 shows the location of the oil separator in a recently equipped plant.

In Fig. 173, there is a check valve in the pipe line connecting the oil separator to the condenser. The purpose of this check valve is to prevent the liquid refrigerant from passing back from the condenser into the compressor, as might happen if it were not removed from the condenser rapidly enough to prevent flooding. If liquid ammonia should flow back into the cylinder

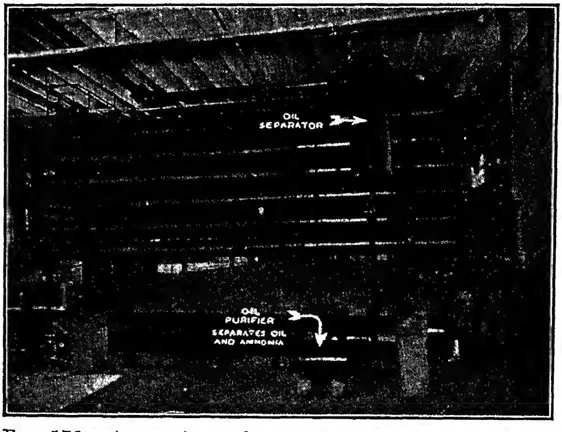


Fig. 176.—Ammonia condenser, oil separator, and purifier.

of the compressor, there would be danger of damaging the cylinder head.

Testing Ammonia Compression Refrigerating Systems for Leaks.—When a new refrigerating compressor is installed, the operating engineer should obtain a set of instructions from the manufacturer, for each machine has its special features and requires special directions for operating, testing, and charging. Some general information, however, may be given which applies to any of the standard types.

[1] The objection to oil separators with baffle plates is that they cause considerable loss of pressure by the sudden change of direction of the flow of the vapor. Such a loss of pressure is the result of the change of kinetic energy into useless heat and, therefore, is an "irreversible" heat process.

Every new plant, when it is installed, should be thoroughly tested before going into use. It is possible that machinery leaving the factory in perfect condition may not be received in such condition. The handling in shipping may cause breakage or inaccurate adjustment.

The most important test is for leaks. This test should be made before admitting any ammonia into the system. The first step is to open the suction pipe at the compressor and to seal it either by screwing a cap over it, or by closing the main suction valve on the end of the pipe. Then permit a free circulation of air through the whole system by opening the main discharge valve, the expansion valve, and any other valves which would interfere with the flow, being careful, however, to shut off the suction-pressure gage, for the operation of testing is likely to damage it. Now start the compressor gradually and let it run slowly. In this way, air passes into the compressor cylinder through the suction valve, is compressed, and is forced through the system. When the high-pressure gage indicates a pressure of 200 to 250 pounds per square inch, stop the compressor and close the main discharge valve tightly. For a few moments, while the air cools, there will be a falling off of the pressure. When the air has cooled, the needle of the gage should remain stationary, *indicating that there are no leaks* and that the system is tight.

If the gage shows a continued falling off in pressure, there are leaks. To locate them, a thick lather of soap and water should be applied to all the piping with a broad flat brush. At the point or points of leakage, the lather will be expanded into soap bubbles by the escaping air. Submerged parts of the system, if there are any, such as the condenser or brine tank, may also have leaky coils. Escaping air in these submerged parts is detected by a column of bubbles, which forms on the coils and rises to the top of the water or brine.

Two methods are effective for repairing small leaks. One is to solder the defective portion of the pipe. The other is to apply a paste of glycerin and litharge, protecting the paste, until it hardens, by binding over it a sheet of rubber. Large leaks cannot be repaired by these means, and the leaky section of pipe must be removed and replaced by a perfect one.

After testing with compressed air, a new plant is sometimes given a vacuum test. To do this, the suction pipe is connected

as for normal operation, and the discharge pipe is disconnected. The valve or cock on the suction gage is now opened, and the one on the discharge or pressure gage is closed. The compressor is operated slowly, as before. The air which was in the system will be sucked into the compressor cylinder and will pass out through the discharge valve. Thus, as the air is pumped out of the system, the suction gage will drop and indicate a vacuum. The main suction valve should then be closed to seal the system. If the suction gage holds its readings, the system is free from leaks.

Charging an Ammonia Compression Refrigerating System.— In charging an ammonia refrigerating system, the compressor must first be operated with the discharge pipe disconnected so that the discharge is into the atmosphere till a vacuum of at least 26 inches of mercury is indicated by the suction gage. Then the discharge pipe is to be attached to the connection for it on the compressor cylinder. After weighing and recording the weight of the ammonia shipping drum, which usually contains about 100 pounds, the ammonia drum should be connected to the charging valve, and the valve marked *A*, in Fig. 173, which is between the liquid receiver and the charging valve on the main liquid ammonia line, should be closed. In this way, the ammonia in the liquid receiver is kept from passing into the cooling coils of the evaporator, and the ammonia in the shipping drum is allowed to expand into the evaporator from which ammonia vapor is drawn into the compressor. The main discharge and suction valves on the discharge and suction pipes at the compressor should then be opened as well as also the expansion valve and the valve at the high-pressure gage. The system is ready to be put into operation when the cooling water begins to circulate in the coils of the condenser. The ammonia vapor can then be drawn from the cooling coils of the evaporator, passed into the compressor, where it is compressed, discharged into the condenser, where it is condensed, and, finally, drained into the liquid receiver.

The piping connections for charging an ammonia refrigerating system are shown in Fig. 177.

Some precautions are necessary in connecting the ammonia shipping drum to the system. A connection of 3/8-inch pipe should be made and bent suitably for connecting the drum to the charging valve of the system. The ammonia drum should be placed and blocked so that the back end is higher than the valve

end, and the outlet valve should point upward. After a tight connection has been made between the ammonia drum and the charging valve, the compressor should be started and operated very slowly. At the same time, the flow of water should be started through its water jacket, if the compressor has one, because a compressor heats very quickly.

The next operation is to open, first, the charging valve and then the valve on the ammonia drum. The valve on the drum must be opened very cautiously, a little at a time. If the odor of ammonia is strong, this valve should be shut off, and the connections to the drum should be tested for leaks. When the

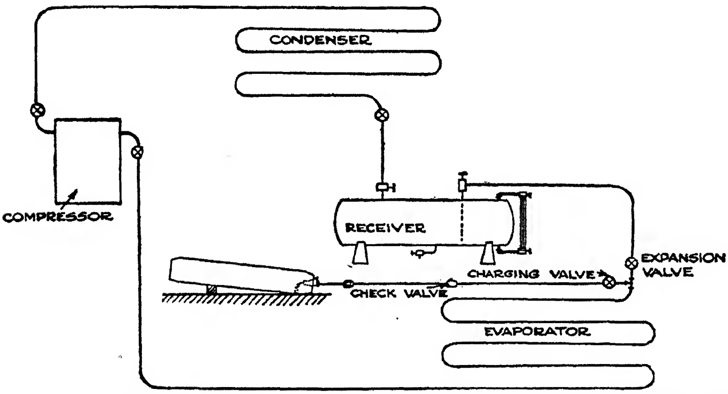


FIG. 177.—Pipe connections for charging ammonia compression refrigerating system.

connections are tight, the valve on the drum may be opened wider.

Soon after the valve on the drum is opened, a coating of frost begins to accumulate on the cooling coils of the evaporator, and gradually this coating of frost extends backward to the charging valve and to its connection to the drum. This frost accumulates because the pressure in the coils of the evaporator is low while the pressure in the drum is high, so that when the charging valve is opened, the liquid ammonia vaporizes at the low pressure in the charging pipe. The evaporation of the ammonia reduces the temperature so much that the moisture of the air settles on the pipe and freezes.

While this part of the charging operation is going on, the compressor is running and discharging the ammonia vapor into the

condenser as soon as it forms. If it were not for this rapid removal of the ammonia vapor from the coils of the evaporator, a high pressure would be produced in them.

When the frost begins to approach the back end of the drum, it indicates that nearly all the ammonia has been removed from the drum. After a while, the frost on the charging pipe begins to disappear. If a blowtorch is applied near the outlet of the drum at this time, the rate of evaporation will be increased, and the drum will be emptied more rapidly.

After all the frost has disappeared from the drum and its connections, the compressor should be stopped, and the suction-gage pressure should be carefully observed. If, after a few moments, there is no appreciable rise in pressure, the drum is empty and may be disconnected. If, however, at this time, the suction-gage pressure rises to atmospheric, the drum is not empty.

In the operation of disconnecting the empty drum, the charging valve should first be closed, and then the valve on the drum. When breaking the connection, one should work slowly and keep one's hands, if not protected, as far as possible from joints, because it is likely that the connections still contain some liquid ammonia which causes burns if touched with bare hands.

If a newly installed system is being charged for the first time, not all the ammonia which will be needed should be put into the system at one time. It is best to charge the system with one-half the necessary amount of ammonia and operate the plant long enough to circulate it throughout the system. Air will collect at the top of the condenser; and this air should be drawn off through the blowoff or "purge" valves, before charging with more ammonia. The ammonia which is still needed may then be added preferably at two different times, circulating the ammonia and drawing off the air through the blowoff or "purge" valves of the condenser between times of filling with ammonia.

When an ammonia drum appears to be empty, it should be weighed, and the weight of ammonia which has been taken out should be checked with that of a full drum. This checking insures fair weight by the dealer and makes certain that the drum is entirely empty. In case more than one ammonia drum is needed, another drum may be attached with the same precautions as before. Before opening the valve of an ammonia charging drum, one should observe that the suction gage of the compressor indicates a vacuum.

After the system has been charged with ammonia, leaks may occur. These are easily detected either by the smell of ammonia or by means of sulphur sticks.

Quantity of Ammonia to Charge System.—The amount of ammonia required for the initial charge is not easy to estimate as one cannot determine the exact quantity of ammonia in each part of the system. The type of condenser, the amount of superheat in the vapor discharged from the compressor, whether or not the coils are flooded or a brine cooler is used, as well as the condition of the suction vapor will affect the amount of refrigerant needed.

The data given in the following table have been used by engineers for estimating the amount of ammonia needed, in pounds per linear foot of pipe in the coils of the evaporator.

	Pound
1¼-inch pipe submerged in water or brine	1/7
1¼-inch pipe exposed to air	1/10
1½-inch pipe exposed to air	1/6
2-inch pipe exposed to air	1/3

To the above weight of ammonia thus determined there must be added the following weights in pounds per stand of pipe (p. 41) to charge the high-pressure side of the system.

	Pounds
2-inch flooded atmospheric type, 12 pipes high by 20 feet long, per stand of pipe	140
2-inch standard atmospheric type, 24 pipes high by 20 feet long, per stand of pipe	60
1¼ × 2-inch double pipe, 8 pipes high by 20 feet long, per stand of pipe	20
1¼ × 2-inch double pipe, 10 pipes high by 20 feet long, per stand of pipe	25
1¼- × 2-inch double pipe, 12 pieces high by 20 feet long, per stand of pipe	30

In the case of ice plants about 45 pounds of ammonia per ton of ice-making capacity may be used for calculating the amount to charge the system. The liquid receiver should always be about half full of the refrigerant.

Removing Refrigerant from the System.—The refrigerant should not be withdrawn from a refrigerating system and placed in shipping drums without weighing the amount of refrigerant placed in each drum. There is great danger of overfilling a

drum. The proper connections for removing the refrigerant from the system are shown in Fig. 178.

Weight of Liquid Ammonia Evaporated in Passing through Expansion Valve.—The total amount of heat which a pound of liquid ammonia will take up in the expansion coil of the evaporator will depend only on the temperature of the liquid ammonia before it enters the evaporator, provided, of course, that all the ammonia is in the liquid form. Now, it happens, always, that some of the liquid ammonia is evaporated when the ammonia in the liquid form passes through the expansion valve, and whatever proportion of a pound of ammonia is in the *vapor form* when

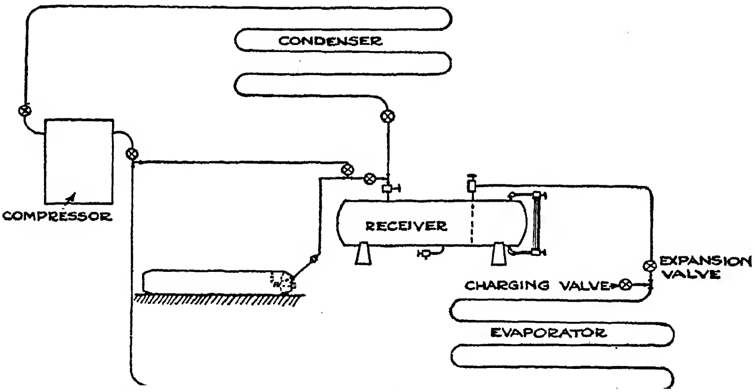


Fig. 178.—Pipe connections for removing ammonia from compression refrigerating system.

it reaches the expansion coil of the evaporator cannot be used for cooling by the method of expansion in the evaporator.

In most cases, in the practical operation of refrigerating plants, the temperature of the liquid ammonia as it enters the expansion coil of the evaporator is nearly the same as its temperature when it leaves the condenser; and this temperature is usually a number of degrees above the temperature of the evaporating ammonia in the expansion coil. Because of this higher temperature of the liquid ammonia, it is necessary to evaporate part of the liquid ammonia in order to cool the remainder from the temperature at which it leaves the condenser to the temperature existing in the expansion coil of the evaporator. This means that just before the liquid ammonia comes to the expansion valve, it is entirely in the liquid state but that immediately after it

passes through the expansion valve and is on its way to the expansion coil of the evaporator, it is a mixture of liquid and vapor.

As an example illustrating this point, it will be assumed that the temperatures in the system are of the usual *standard values*. That is, the temperature of the ammonia vapor entering the condenser is 86° F., and the temperature in the expansion coil of the evaporator is 5° F. The ammonia must, therefore, be cooled in its passage through the condenser and through the expansion valve from 86° F. to the lower temperature or through a range of 81° F. The temperature of the ammonia vapor, when becoming liquid in the condenser, is reduced from 86° F. to nearly the temperature of the water used for cooling the condenser; and the rest of the temperature reduction must come about by the evaporation of part of the liquid ammonia. The heat required to cool the liquid ammonia from 86 to 5° F. may be found by subtracting from the total heat in the liquid ammonia at 86° F. the total heat in the liquid ammonia at 5° F. These values of total heat of the liquid ammonia may be taken from Table of Properties of Ammonia in the Appendix; in this table, it is found that the total heat of the liquid at 86° F. is 138.9 B.t.u. per pound and that the total heat of the liquid ammonia at 5° F. is 48.3 B.t.u. per pound. The difference of the total heats of the liquid is, therefore, 90.6 B.t.u. per pound. For purposes of illustration, it may now be assumed that the ammonia is cooled by means of very cold circulating water to a temperature of 66° F. A certain amount of every pound of liquid ammonia must, therefore, be cooled from 66 to 5° F. (temperature in the evaporator). The total heat in the liquid ammonia at 66° F. is 116.0 B.t.u. per pound, and the total heat of the liquid ammonia at 5° F. is 48.3 B.t.u. per pound. The difference or 67.7 B.t.u. per pound must, therefore, be absorbed from the liquid ammonia in order to cool it from the temperature at which it leaves the condenser (66° F.) to the temperature in the evaporator.

The above calculations show that when cold water is available for use in the ammonia condenser, the cooling effect theoretically available in the expansion coil of the evaporator is 565.0 B.t.u. per pound (latent heat of evaporation at 5° F.) less 67.7 B.t.u. per pound, or 497.3 B.t.u. per pound. On the other hand, if the cooling water which is available is at a relatively high tem-

perature, so that the liquid ammonia leaves the condenser, at 86° F., the cooling effect in the expansion coil of the evaporator due to each pound of ammonia is 565.0 − 90.6 or only 474.4 B.t.u. per pound.

Percentage of Ammonia Evaporated by Cooling between Condenser and Evaporator.—The amount of ammonia which must be evaporated in order to reduce the temperature between the condenser and the evaporator is easily expressed as a *percentage.* In other words, this is a percentage loss of refrigerating effect due to this reduction in temperature. When the liquid ammonia leaves the condenser at 66° F., the percentage by weight of ammonia which must evaporate so that its temperature as it enters the expansion coil of the evaporator may be 5° F. is 67.7 ÷ 565.0 = 0.12, or 12 per cent.

Now, if the liquid ammonia leaves the condenser at 86° F., with other conditions the same, the percentage of each pound of ammonia evaporated in the cooling between the condenser and the evaporator is 90.6 ÷ 565.0 = 0.16 or 16 per cent. In each case, the latent heat of evaporation is, of course, 565.0 B.t.u. per pound.

This percentage of ammonia which must be evaporated to provide for the cooling between the condenser and the evaporator is really the percentage of "wetness" of the ammonia vapor after it has passed the expansion valve and enters the expansion coil. The *quality* of a vapor is usually expressed as a decimal and is the difference between unity and the decimal fraction corresponding to the percentage of wetness. In the first case above, when the liquid ammonia leaves the condenser at 66° F., the *quality* of the ammonia vapor entering the expansion coil of the evaporator is 1.00 − 0.12 or 0.88.

Effects of Cooling-water Temperature on Refrigerating Effect.—A comparison of the cooling effects when ammonia leaves the condenser at 66 and at 86° F. shows the desirability of cooling the liquid ammonia to a temperature as low as possible in the condenser. In other words, it shows strikingly the improved refrigerating effect from using cold water for cooling the condenser.

Testing for Leakage of Ammonia Vapor.—In the regular operation of an ammonia refrigerating plant operated with compressors, those in charge of the machinery must be constantly on the lookout for the leakage of ammonia vapor through the piston-rod

packing and valve stems. In all kinds of plants using ammonia, there is the danger of leakage at the joints in the piping. It is very difficult to keep the packing around the piston rod perfectly tight at all times, and tightness becomes especially difficult to maintain when the piston rod is not centrally located at all times in the stroke. One reason for this difficulty is that the piston rod will eventually wear away more in the middle of the stroke than at the ends, and this unequal wear may become so great that it will be necessary to turn down a rod in a lathe. If the wear in the crosshead shoe is not adjusted carefully from time to time, there will be present another cause for excessive wear on the piston-rod packing.[1]

The test most commonly used to determine whether or not there is leakage of ammonia vapor through the piston-rod packing, valve stems, and pipe joints is to burn a stick of sulphur near the place where leakage is suspected. A leak of ammonia vapor will be indicated by white fumes, as already stated on page 77. Condensers of the atmospheric and shell-and-tube types may also be tested with sulphur sticks, when the tested surfaces are dry and the gage pressure in the condenser is about 200 pounds per square inch.

A double-pipe condenser may be tested for leaks by the use of sulphur sticks, but, obviously, the method of testing with sulphur will not work out so well with this type of condenser as with other types and under other conditions. For this reason, a double-pipe condenser is usually tested for leaks by adding what is called *Nessler's solution* to a sample of the cooling water which is discharged from the condenser while it is operating. In order to make the test, a sample of the cooling water should be collected in any kind of glass vessel, and, if any ammonia is present in the sample, the water will be colored when a few drops of Nessler's solution are added. It will turn yellow, if relatively small leaks are present, and dark brown, if the leaks are large.

Thus, large leaks of ammonia from the expansion coil can be tested by Nessler's solution added to the brine, and the brine in the freezing tank of an ice plant can also be tested in the same way. If the brine has been made by the addition of sodium chloride to water, Nessler's solution can be added to the brine in

[1] Records of leakage of ammonia in refrigerating plants in Philadelphia and Chicago show that the average loss of ammonia by leakage in well-operated plants adds an expense of about 2 cents per ton of ice.

the same manner as it would be put into the cooling water of the condenser. On the other hand, if the usual kinds of calcium brine are to be tested for ammonia leakage, it is necessary to remove the calcium by adding to the sample of brine after dilution with water enough sodium carbonate to precipitate the calcium and then adding Nessler's solution to the filtrate. If the filtrate turns brown after Nessler's solution has been added, there is a leakage of ammonia into the sample being tested. The addition of sodium to the calcium brine before testing with Nessler's solution is necessary, because Nessler's solution always forms a precipitate in calcium brine whether or not ammonia is present. It will be observed, however, that if there is no leakage of ammonia into the brine, the sample being tested will be white when the Nessler's solution is added, and will have a yellow color when only a trace of ammonia is present.

Testing with Nessler's Solution and Litmus Paper.—The formula for Nessler's solution is as follows: Dissolve 17 grams of mercuric chloride in about 300 cubic centimeters of distilled water; dissolve 35 grams of potassium iodine in 100 cubic centimeters of water. Add the former solution to the latter, with constant stirring, until a slight permanent red precipitate is formed. Next, dissolve 120 grams of potassium hydrate in about 200 cubic centimeters of water; allow the solution to cool, and then add it to the previous solution and make up with water to 1 liter. Add mercuric-chloride solution until a permanent precipitate again forms. Allow to stand till settled, and decant the clear solution for use. Put it in glass-stoppered *blue* bottles and set aside in a dark place in order to prevent decomposition.

Litmus paper, of the kind which can be purchased in any drug store for testing alkaline reactions, can be used to test water or brine for the presence of ammonia. A portion of a strip of this kind of prepared paper is simply dipped into the sample of water or brine to be tested. If ammonia is present, the moist part of the paper will turn blue. Small amounts of ammonia leakage will change the color of the paper scarcely at all, while a large amount of leakage will turn it to a deep blue. Litmus paper will not, however, give reliable tests of calcium brine, as a strong calcium brine will turn it blue in the same way that ammonia does.

Stick Test to Check Operating Conditions.—The liquid receiver should always be provided with an adequate gage glass, so that

the depth of liquid can always be readily determined. The gage glass should be protected with four wires, as shown in Fig. 179, and should also have automatic safety valves, like those in Fig. 180, which will shut off the connections if the glass is broken. The level of the liquid in the receiver is always changing, and if a plant is being operated with more than one temperature in a group of expansion coils, there is likely to be continual trouble in maintaining the capacity of the plant unless the person in charge can tell at any moment how much liquid ammonia there is in the system back of the expansion valve. If it happens

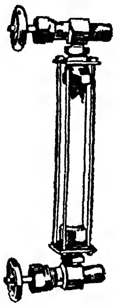


FIG. 179.—Metal guards for gage glass.

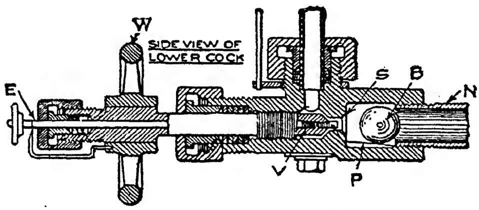


FIG. 180.—Details of safety gage

that the level in the liquid receiver is drawn down so much that there is no supply, then liquid ammonia and uncondensed gas will flow as a mixture into the coils of the evaporator, where the uncondensed gases will have the effect of choking the coils and thus reducing the refrigeration capacity.

In this connection, it is convenient to have some means of determining whether or not the operating conditions in the coil of the evaporator are satisfactory. The *stick test* is often used to determine whether or not the proper temperature is being carried in the coils of the evaporator. The method of testing is to scrape away the frost on a pipe of the coil to be investigated and then to apply a moistened finger to the surface of the pipe. If the temperature in the pipe is below 15° F., the finger will stick to the surface of the pipe, and a considerable pull will be required

to remove it. On the other hand, if the finger does not stick to the pipe, it shows that the necessary low temperature is not maintained.

Draining Oil from the Oil Separator and Liquid Receiver.—Figure 173 shows the *liquid receiver* and the *oil separator* placed side by side. This is the usual arrangement, and, in most cases, they are connected by an *oil drum,* which is located beneath them. A small connection between the main suction pipe and this oil drum is also usual.

In order to drain the oil separator, the valve between the separator and the oil drum is opened, and, after allowing ½ hour for the oil to flow into the drum, this valve is closed, and the one between the oil drum and the main suction pipe is opened. The best results are obtained when this latter valve is opened only a little. When white frost gathers on the connection between the main suction pipe and the oil drum, there is an indication that the ammonia which was mixed with the oil is being drawn off into the suction pipe. After this frost has disappeared, the valve may be closed, and then, after waiting 20 minutes for the oil to settle in the oil drum, it may be drawn off into a bucket by opening the drain valve at the bottom of the oil drum.

Since oil collects only very slowly in the liquid-ammonia receiver, it will not be necessary to draw it off so frequently as from the oil separator. The method of drawing oil from the liquid-ammonia receiver is similar to that for draining the oil separator.

The Bypass.—Bypass pipes are provided in most compressors for the purpose of changing the suction pipe into a discharge pipe or the discharge pipe into a suction pipe. It consists of two connections each with a valve, one leading from the discharge pipe to the suction pipe, and the other from the suction pipe to the discharge pipe. Figure 174 shows a common arrangement of the *bypass valves,* as marked at A and B. The *main discharge valve* is lettered D, and the main suction valve S. If D and S are closed and A and B are opened, ammonia will discharge through the small bypass connection *from* the discharge side of the compressor *into* the main suction line. Through the other bypass connection, ammonia will be drawn from the main discharge line into the suction pipe of the compressor.

Pumping Out the Condenser of an Ammonia Refrigerating System.—Whenever repairs or cleaning make it necessary to

open the condenser, the ammonia must be removed by pumping through the bypass pipes (see p. 210). In the process of removing the ammonia from the condenser, the valve at the condenser on the liquid-ammonia line between the condenser and the liquid-ammonia receiver should now be closed, in order to shut off that part of the system containing the main discharge line, oil separator, and condenser from the part containing the main suction line, cooling coils of the evaporator, main liquid line, and liquid receiver. It is a good plan to trace out these parts on Fig. 173. The main discharge and suction valves should then be closed, and both bypass valves should be opened. The system is now to be reversed, so that the compressor draws the ammonia from the condenser through the discharge line and forces it into the opposite side of the system (through the main suction line) into the cooling coils of the evaporator. For this transfer of ammonia, the compressor must be operated at very low speed, and the expansion valve should be wide open. The ammonia enters the cooling coils of the evaporator at high pressure and, consequently, also at high temperature and will be condensed. To operate the compressor at high speed would be likely to damage the system, since the compressor is now discharging high-pressure ammonia vapor through a small pipe.

When the discharge-pressure gage indicates that a vacuum has been established in the condenser, the two bypass valves should be closed tightly, and the work of repairing the condenser may begin. It will also be possible to open any other parts *on the same side* of the system. The opening of any part of the system in which a vacuum has been established admits air into the system and destroys the vacuum. Before more ammonia is admitted into this side of the system, the air must be removed. Provision is made for removing air from the system shown in Fig. 174 by the valve Z, which has a blank flange instead of the connection Y. To establish a vacuum by means of this valve, the main discharge and suction valves D and S, respectively, are closed, and the bypass valve at A is opened, the blank flange is removed, and a pipe is attached to the valve Z as shown at Y. Because the air which is drawn off may probably contain strong ammonia fumes, it is a good idea to run the free end of the pipe outside the room so that the discharge will not foul the inside air. Having opened the valve Z, the compressor should be started at a very low speed, and the air is removed from the con-

denser through the bypass of the compressor into the main discharge pipe and is expelled through the pipe leading from *Z*. When this side of the system is cleared of air, the discharge-pressure gage will indicate the presence of a vacuum. Now the valves *Z* and *A* may be closed, the blank flange at *Y* replaced, and the main discharge valve *D* opened.

To put the system once more in regular operation, the flow of condenser cooling water should be started, and the main suction valve *S* should be opened very slowly, all the usual precautions for starting the compressor being observed.

Removing Ammonia from Evaporator.—The method of removing ammonia from the cooling coils of the evaporator is somewhat different and does not require the use of the bypass pipes. First, in order to prevent any more ammonia from passing into the coils of the evaporator, the main liquid line valve (marked *A*, in Fig. 173) is closed. The compressor is then operated as in regular service, allowing the flow of condenser cooling water to continue. After a time, the suction-pressure gage will indicate a vacuum, and the frost will leave the suction pipe, showing that the ammonia has been pumped through the condenser into the liquid-ammonia receiver. When the compressor has been stopped and both the main discharge and suction valves have been closed, the suction side of the system may be opened for repairing.

In emptying the oil separator or the liquid-ammonia receiver for repairs, the compressor should be stopped, and the liquid ammonia in the receiver should be emptied into the cooling coils of the evaporator. When this has been done, the valve on the main liquid line (*A*, in Fig. 173) should be closed, and the ammonia should be removed in the same way as it is taken from the condenser, that is, the remaining ammonia should be discharged (by using the bypass) through the compressor into the cooling coils of the evaporator, taking care to operate the compressor very slowly. The ammonia is thus stored in the coils of the evaporator while the repairs are being made.

Operating Practice for Absorption System.—Under ordinary conditions, in plants having *low pressures* in the absorber and evaporator, it is probably best to pass all the cooling water through the absorber first and then through the condenser,[1]

[1] The statement is generally made that the temperature of the cooling water going to the condenser should be as low as possible. In the case of low pressures (and low temperatures) in the absorber and evaporator, there

after which the water may be used in the rectifier or weak-liquor cooler. In *high-pressure* plants, it may be desirable to pass all of the water through the condenser and then all or part through the absorber, after which it may then be used in the rectifier and in the weak-liquor cooler.

Operating Data of Absorption System.—In the following example, relating to an absorption refrigerating system, the strengths of solutions, temperatures, and pressures are not actual values from tests but are given here merely to illustrate what takes place in the different parts of the system.

Referring now to Fig. 17, it will be assumed that the strong liquor entering the generator has 29 per cent of ammonia and that its absolute pressure is 175 pounds per square inch. The steam applied to the heating coils of the generator has an absolute pressure of 65 pounds[1] per square inch, the temperature corresponding to this pressure being 298° F. The liquor in the generator will then be heated by the steam to a temperature of about 270° F.

If, for example, ammonia vapor containing 10 per cent of water vapor leaves the generator at a temperature of 270° F. and rises through a rain of strong liquor, in the analyzer, there is an exchange of heat, and the strong liquid is heated by the hot vapors. Strong liquor from the exchanger, having a temperature of, say, 210° F. and a strength of about 29 per cent, enters the top of the analyzer, as indicated in the figure. This strong liquor then flows over the edges of the pans and falls from one to the other until it passes down into the top of the generator. If the temperature of the strong liquor, when entering the analyzer, is 210° F. and, as a result of the transfer of heat, the temperature is raised to 225° F., some of the ammonia will be driven out of the strong-liquor solution. This ammonia vapor then passes on with the hot vapors from the generator through the horizontal

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is not the necessity, in most plants, for having the lowest possible temperature of the cooling water going to the condenser.

[1] The boiler should generate steam at an absolute pressure somewhat greater than 65 pounds per square inch. In case the pump is driven by steam, this pressure at the steam cylinder of the pump should be high enough so that the exhaust from the pump can be used in the steam coils of the generator at an absolute pressure of 65 pounds per square inch. In case extra steam should be needed for the generator steam coil, it can be supplied from the boiler through a throttling valve, which will reduce the boiler pressure to the desired pressure for the steam coils in the generator.

pipe connecting the analyzer and the rectifier. The strong liquor entering the generator from the analyzer has now a temperature of about 225° F. and a strength of about 28 per cent of ammonia. On reaching the generator, it is again heated by the steam coils to a temperature of 270° F., and its strength is reduced to about 21 per cent of ammonia. It then leaves the generator as weak liquor at a temperature of 270° F. and a strength of 21 per cent of ammonia.

The vapors rising from the generator and passing up into the analyzer are at a temperature of 270° F. and contain some water vapor. This water vapor is cooled in the analyzer by the strong liquor, so that some of it is condensed, and the condensation returns to the generator. The remaining vapor, containing about 7 per cent of water vapor, is cooled to about 235° F. and passes out of the analyzer into the rectifier.

If the mixture of ammonia and water vapors enters the rectifier at a temperature of 235° F. and the strong liquor from the absorber enters the rectifier at a temperature of 110° F., in the exchange of heat the ammonia vapor will be cooled to about 135° F. At this temperature, it passes to the condenser at a gage pressure of 165 pounds per square inch. The strong liquor is heated to a temperature of 150° F. in passing through the rectifier. From here the strong liquor then passes into the exchanger.

The exchanger, shown in Fig. 17, is of the double-pipe type (p. 27). It transfers heat from the hot weak liquor to the cool strong liquor. The weak liquor entering the exchanger has a temperature of 270° F., while the entering strong liquor has a temperature of 150° F. The temperature of the strong liquor is then increased to about 210° F. and passes out of the exchanger into the analyzer.

The weak liquor, after being cooled in the exchanger, enters the weak-liquor cooler. This cooler is also of the double-pipe type. In this cooler, the weak liquor gives up heat to the cooling water supplied from the condenser. It is then cooled from 175° F. to the temperature of the absorber, which is about 110° F. The cooling water leaving the condenser has a temperature of about 85 to 89° F. As a large quantity is available, it will cool the weak liquor through a large range of temperature without raising its temperature very much.

After leaving the cooler, the weak liquor passes through a regulating valve. This valve reduces the gage pressure from 165

pounds per square inch to 15 pounds per square inch, this being the pressure of the anhydrous-ammonia vapor in the evaporating coils of the brine cooler. The regulating valve is generally of the automatic type, and it adjusts itself so as to maintain a constant pressure of the weak liquor entering the absorber.

After the water vapor has been removed from the ammonia vapor in the rectifier, the ammonia vapor passes over into the condenser at a gage pressure of 165 pounds per square inch and a temperature of 135° F. Owing to the fact that it is superheated, it must first be cooled to about 89° F. before it will condense. After it is condensed, it may be cooled a few degrees more in the condenser, so that it will enter the liquid receiver at a temperature of about 85 to 87° F. From the liquid receiver, the liquid ammonia flows through the expansion valve into the coil of the evaporator in the brine cooler. In the coil of the evaporator, it evaporates at a gage pressure of about 15 pounds per square inch and a temperature of 0° F. The resulting ammonia vapor then passes into the absorber, where it mixes with the weak liquor from the weak-liquor cooler. The weak liquor, now having a strength of 21 per cent, a temperature of 125° F., and a gage pressure of 15 pounds per square inch, absorbs the ammonia vapor. This absorption of ammonia increases its strength to 29 per cent at 110° F. The resulting strong liquor is then pumped into the rectifier again.

Heat is generated when the absorption of ammonia vapor takes place. This excess heat must be carried away, so that the weak liquor may absorb a large quantity of ammonia vapor, thus producing a stronger liquor. This excess heat is often carried away by the cooling water, which has already been used in the condenser and now passes through the cooling coils in the absorber.

In some plants, the cooling water from the condenser is not used to cool the weak liquor as shown in Fig. 17, for the reason that since this condenser cooling water has already acquired a great deal of heat, it would not be very effective in cooling the absorber. In this case, an independent water supply for the absorber is used.

The strengths of the strong and weak liquors are controlled by four conditions: (1) absorber pressure, (2) generator pressure, (3) steam pressure in the generator coils, (4) speed of the liquor pump. In order to obtain the best results, this system must be operated with strengths of strong and weak liquor that will give

1 pound of liquid anhydrous ammonia for every 7 to 8 pounds of strong liquor. In order to do this, the strength of the strong liquor should be as strong as possible, to correspond with the pressure and temperature in the absorber. On the other hand, the weak liquor should be as weak as possible. The temperature and pressure in the generator are generally fixed by the temperature of the cooling water for the condenser. The only remaining variables to adjust are the steam pressure in the generator coils and the speed of the pump.

Before the system can be operated satisfactorily, it will be necessary to equip it with pressure gages as follows: (1) steam pressure gage for the generator coils; (2) ammonia pressure gages for the generator, condenser, evaporating coils, and absorber. Also, there should be provided gage glasses (water gages) on all apparatus where it is necessary to know the amount of the liquid contained. These may be provided with self-closing safety cocks.

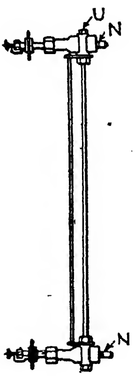

FIG. 181.—Safety gage glass.

A ball type of safety water gage is shown in Figs. 180 and 181. The complete gage is shown in Fig. 181 and a section of the lower gage cock is shown in Fig. 180. The gage cocks are connected to the receiver or drum, where they are to be used by means of extra-heavy nipples *N*, which should be inserted into the cock so that the ball *B* may move freely back and forth on the inclined plane *P*. The plug in the upper cock is large enough to permit inserting the gage glass from that end. In case the glass breaks, the pressure of the liquid or vapor forces the balls *B* in the cock, as shown in the figure toward the left against the seats *S*, so that they prevent any flow into the gage glass. The gage cocks at top and bottom should then be closed by turning the handwheels *W* in a clockwise direction. After a gage glass has been replaced, the "unseating" stem *E* should be screwed in far enough so that it will press the ball *B* off the seat *S* before the cocks are opened by the movement of the handwheels *W*. The cocks should be opened slowly, and the balls *B* will then slide back on the inclined planes *P* to the position shown in the figure. An approved method of protecting the gage glass by means of four rods is illustrated in Fig. 179.

A gage glass should not be longer than 2 feet, but, in some cases, where the depth of the liquid to be measured exceeds 2 feet, two gage glasses, one above the other, should be used instead of one.

In a refrigerating system, air is likely to leak in, and some means must be provided to rid the system of it. For this reason, *purge valves* should be connected at all points where air is likely to collect.

Some means must be arranged to *charge the system.* For this purpose, charging valves are provided. The usual place for charging the liquid ammonia in an absorption system is shown in Fig. 17 just below the pump. This valve may be used to drain off the liquor when repairing or for taking a sample. The valve V may be used to obtain a sample of the weak liquor.

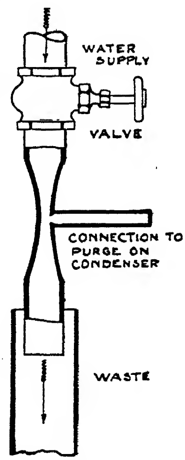

FIG. 182.—Venturi tube for connection to condenser.

Testing a New Absorption System.—The method of testing a new absorption system for tightness is similar to that used in testing a compression system. The system should be subjected to an air pressure of about 250 pounds per square inch (gage) and allowed to stand at this pressure. If the pressure remains constant after the air has cooled, the system is tight, and it should not be necessary to test it with a vacuum.

As a new system will contain air, the latter should be removed before charging the system with liquor. In doing this, the discharge side of the liquor pump should be disconnected. The free end of the system should now be sealed by closing the valve. All other valves of the system should be opened so as to permit the air to pass over to the pump, thus preventing an accumulation of air in a portion of the system after the pumping. The pump should be started for the purpose of removing the air from the system through the absorber, continuing until the vacuum gage shows the highest possible vacuum that can be obtained. The valve on the suction pipe should now be closed, and the discharge end of the pump reconnected. The system is now ready to receive its charge of liquor.

Another way of removing air from the system is by the use of the *venturi tube,* as shown in Fig. 182. This method is the better, as through it a high vacuum can be easily obtained. As

shown in this figure, the venturi tube consists of a short length of pipe which narrows down to a small diameter at the middle and enlarges again to the original diameter of the pipe. A connection is made at the narrow section. When a stream of water is supplied to the tube at the proper pressure, the stream will pass the connection at the narrow section with a very high velocity. Some air is caught by the water and is carried along with it. Thus, there is a removal of air. When removing the air from an absorption system with one of these venturi tubes, the tube should be connected to the water supply of the condenser, and the *purge pipe* (p. 44) *at the top of the condenser* should be joined to the connection at the narrow section of the venturi tube. By running the water through the tube, the air is removed from the system, and a high vacuum is obtained.

Charging an Absorption Refrigerating System.—After a good vacuum has been obtained in the system, it may be charged with liquor. The first step is to connect a shipping drum filled with aqua-ammonia liquor to the charging valve. The charging valve and the valve on the connection of the shipping drum should then be opened. After this is done, the air valve on the shipping drum is also opened. This produces a difference of pressure on the liquor, forcing it into the system. The system should be only partly charged at first. It should then be operated for a time until the remaining air in the system collects in the condenser. This air should then be expelled, after which the remainder of the required charge may be admitted. When the drum is nearly empty, great care should be taken to prevent air from entering the system. If the system is fully charged at first and is then started, the liquor will expand and cause an overcharge. This will flood the condenser and may damage some parts of the system.

When enough liquor has been added so that there is a slight pressure instead of a vacuum in the system, the ammonia pump should be started, steam should be allowed to enter the generator coils, and the condenser cooling water should be started. The liquor should be heated until the gage pressure in the generator is about 100 pounds per square inch. The ammonia vapor will then begin to condense and collect in the liquid receiver. When a sufficient quantity of liquid ammonia has collected, the expansion valve may be opened. The system is now in operation so that the water may be turned on the absorber, the liquor

pump started, the air expelled, and the remaining charge of liquor put in. Enough liquor should now be put into the system so that, when operating under normal conditions, there will be about 1 foot of liquid ammonia in a vertical receiver.

Recharging an Absorption System.—After a season of continuous running, an absorption system will lose some of its ammonia by leakage. Not only is anhydrous ammonia lost in this manner, but there is also a leakage of liquor. Such losses tend to reduce the liquor's strength and make it necessary from time to time to charge the system with new ammonia or liquor. This should be done during the months of the year when refrigeration is not essential. At such times, the plant can be shut down, and charging can be done more conveniently than when it is in operation. If the strength of the charge becomes too low, it can be easily strengthened by adding liquid anhydrous ammonia until the strength is again normal. On the other hand, if the charge has reduced in volume only, more liquor must be added. The amount of liquor in the system can be determined by the gage glasses on the absorber and generator. The strength of the liquor can be determined by taking a sample and testing it with a hydrometer.

Strengthening the Charge of Aqua-ammonia Liquor.—When ammonia is absorbed in water, its specific gravity changes and becomes less than that of water, which is, of course, unity. In order to obtain the specific gravities of liquids, a calibrated instrument called a *hydrometer* is used. It is made of a glass tube and is hollow so that it will float. At one end, there is a bulb partly filled with lead shot or mercury. The hydrometer will float upright when placed in a solution. If it is a direct-reading hydrometer, the number corresponding to the level of the liquid indicates the specific gravity. The stems of hydrometers, however, are often marked with an arbitrary scale of degrees, called the *Baumé scale.* The point on this scale which is marked 10 degrees is the specific gravity of water or 1 on the specific-gravity scale. The Baumé scale is generally used for measuring different strengths of aqua-ammonia liquors.

This relationship may be expressed mathematically by the following equation:

$$\text{Degrees Baumé} = \frac{140}{\text{sp. gr.}} - 130.$$

In addition to the specific gravity of aqua-ammonia solutions,

TABLE Vc.—SPECIFIC GRAVITY OF AQUA AMMONIA

Degrees Baumé	Specific gravity	Ammonia, per cent	Degrees Baumé	Specific gravity	Ammonia, per cent
10.00	1.0000	0.00	19.50	0.9365	16.80
10.25	0.9982	0.40	19.75	0.9349	17.28
10.50	0.9964	0.80	20.00	0.9333	17.76
10.75	0.9947	1.21	20.25	0.9318	18.24
11.00	0.9929	1.62	20.50	0.9302	18.72
11.25	0.9912	2.04	20.75	0.9287	19.20
11.50	0.9894	2.46	21.00	0.9272	19.68
11.75	0.9876	2.88	21.25	0.9256	20.16
12.00	0.9859	3.30	21.50	0.9241	20.64
12.25	0.9842	3.73	21.75	0.9226	21.12
12.50	0.9825	4.16	22.00	0.9211	21.60
12.75	0.9807	4.59	22.25	0.9195	22.08
13.00	0.9790	5.02	22.50	0.9180	22.56
13.25	0.9773	5.45	22.75	0.9165	23.04
13.50	0.9756	5.88	23.00	0.9150	23.52
13.75	0.9739	6.31	23.25	0.9135	24.01
14.00	0.9722	6.74	23.50	0.9121	24.50
14.25	0.9705	7.17	23.75	0.9106	24.99
14.50	0.9689	7.61	24.00	0.9091	25.48
14.75	0.9672	8.05	24.25	0.9076	25.97
15.00	0.9655	8.49	24.50	0.9061	26.46
15.25	0.9639	8.93	24.75	0.9047	26.95
15.50	0.9622	9.38	25.00	0.9032	27.44
15.75	0.9605	9.83	25.25	0.9018	27.93
16.00	0.9589	10.28	25.50	0.9003	28.42
			25.75	0.8989	28.91
16.25	0.9573	10.73	26.00	0.8974	29.40
16.50	0.9556	11.18	26.25	0.8960	29.89
16.75	0.9540	11.64	26.50	0.8946	30.38
17.00	0.9524	12.10	26.75	0.8931	30.87
17.25	0.9508	12.56	27.00	0.8917	31.36
17.50	0.9492	13.02	27.25	0.8903	31.85
17.75	0.9475	13.49	27.50	0.8889	32.34
18.00	0.9459	13.96	27.75	0.8875	32.83
18.25	0.9444	14.43	28.00	0.8861	33.32
18.50	0.9428	14.90	28.25	0.8847	33.81
18.75	0.9412	15.37	28.50	0.8833	34.30
19.00	0.9396	15.84	28.75	0.8819	34.79
19.25	0.9380	16.32	29.00	0.8805	35.28

it is obvious that the percentages of ammonia and water are also important. The percentage of ammonia in solution will also vary the specific gravity. This may be expressed by the following equation, where X is the percentage of ammonia in the mixture, called the percentage of *concentration:*

$$\text{Specific gravity} = 1 - \frac{4.3}{1{,}000}\left(X - \frac{X^2}{100}\right)$$

The relationship between the specific gravity, degrees Baumé, and the percentage concentration may be found from Table V*c*. The specific gravity determinations in the table were made at 60° F. being compared with the specific gravity water at the same temperature. It often happens that the specific gravity is determined at other temperatures than 60° F.; and, since the specific gravities would be affected by the change in volume due to the change of temperature, suitable corrections must be made. The *correction factor* may be found in Table V*d*.

Table V*d*.—Allowances for Temperatures of Aqua-ammonia Solutions

	Corrections to be added when temperature is below 60° F.		Corrections to be subtracted when temperature is above 60° F.		
Degrees Baumé	40° F.	50° F.	70° F.	80° F.	90° F.
14°	0.015° Bé.	0.017° Bé.	0.020° Bé.	0.022° Bé.	0.024° Bé.
16°	0.021° Bé.	0.023° Bé.	0.026° Bé.	0.028° Bé.	0.030° Bé.
18°	0.027° Bé.	0.029° Bé.	0.031° Bé.	0.033° Bé.	0.035° Bé.
20°	0.033° Bé.	0.036° Bé.	0.037° Bé.	0.038° Bé.	0.040° Bé.
22°	0.039° Bé.	0.042° Bé.	0.043° Bé.	0.045° Bé.	0.047° Bé.
26°	0.053° Bé.	0.057° Bé.	0.057° Bé.	0.059° Bé.	

When the charge needs strengthening, a shipping drum containing liquid anhydrous ammonia should be connected to a charging valve on the coils of the evaporator. As it is possible to find the amount of liquor in the system and its strength, the amount of anhydrous ammonia to be added to give the liquor the proper strength can be calculated. Methods are given on page 298.

For charging with anhydrous ammonia, the shipping drum should be placed on platform scales and connected to the charging

valve by means of a flexible connection. The weight of the drum should be noted. While charging, the scales should be kept balanced, in order to determine the amount of ammonia that has passed into the system. When the system has received all but 10 to 15 pounds of its new charge, the charging valve should be closed, and the system put in operation for 2 or 3 hours. This permits the new anhydrous ammonia to become thoroughly absorbed in the liquor. The remainder can then be put into the system. If the charge has become both weakened and reduced in amount, it is best to charge the system with liquor having a strength of about 26° Bé. This liquor should be added until the strength is normal.

In charging the system with liquor, the shipping drum should be connected to the charging valve, and the drum should be raised above the liquor pump, as otherwise the pump is likely to "race." This is because the suction pressure is reduced and the pump has less resistance then than when handling liquor. The vapor thus formed in the cylinder of the pump can then be condensed by pouring cold water upon the cylinder, which will sometimes stop this trouble. Racing can be entirely prevented by simply raising the charging drum well above the pump.

Leaks in an Absorption System.—The absorption system is more likely to have leaks than is the compression system, as considerable pitting or corrosion occurs. The generator and the exchanger coils are especially subjected to corrosion. Because of this, a constant watch must be kept, and all leaks given immediate attention. All of the generator coils should be kept well covered with liquor, as corrosion takes place more rapidly when the coils are uncovered than when covered. These coils should be carefully inspected when the plant is shut down, to make sure that they are in good condition.

In plants equipped with an analyzer, a mixture of ammonia and steam is formed in the generator and enters the analyzer. This water vapor causes corrosion and leaks in the analyzer pans. Some water may even collect in the liquid receiver. A sample should be taken occasionally from the receiver and tested by evaporating it in a glass test tube. If the water remaining is as much as 20 per cent by volume of the original sample, the analyzer pans are probably leaking.

When leaks occur in the exchanger, there results a mixing of the strong and weak liquor. This changes the strengths of these

liquors, the strong liquor becoming weaker and the weak liquor becoming stronger. When such a leak is large, it may be detected, in extremely bad cases, by the cooling of the weak-liquor pipe between the exchanger and the generator. The cooling of this pipe is due to the fact that the strong liquor is pumped through the leak into the weak-liquor connections, so that the former enters the generator at the top and bottom.[1] A leak in the exchanger may also be detected by closing the valve in the weak-liquor line from the generator to the exchanger. The valve in the weak-liquor feed line to the absorber should first be closed. The liquor pump should be started very slowly, and then the valve in the weak-liquor feed line should be opened wide. If the pump speeds up, there is a serious leak in the exchanger. The reason for this is that when the valves in the weak-liquor feed line to the absorber and in the weak-liquor line from the generator to exchanger are both closed, the pump will be doing work against the pressure of the generator. Now, if there is a leak in the exchanger, and the valve in the weak-liquor feed line to the absorber is opened, the pump will work against a lower pressure. This is because the strong liquor is passing through the short circuit made up of the exchanger, the leak, and the weak-liquor cooler. From the weak-liquor cooler, the liquor returns to the absorber.

Boil-overs.—By *boil-over* is meant the result of liquor's entering some part of the system in which it does not belong. This is generally caused by the accumulation of condensed steam in the steam coils of the generator. This accumulation of water reduces the active steam surface. In attempting to maintain the desired generator pressure, the engineer turns on more live steam, to make up for the reduced heating surface of the steam coils. Now, if the condensation in the steam coils should stop, the liquor will receive too much heat and boil over into the condenser. In order to prevent this, a steam trap should be connected to the steam coils. This trap will drain the steam coils without loss of steam.

A boil-over sometimes occurs as a result of the ammonia's being low in the system. The liquor is then likely to be syphoned over from the generator into the absorber. This is the result of too rapid absorption of ammonia. For this reason, the gage glass on

[1] In such case, there will be an increase in pressure in the generator, with, probably, the stopping of refrigeration.

the generator should be frequently observed. By so observing, it is easy to determine when the level of the liquor in the generator is getting too low.

Effects of Air on the Refrigerating System.—There is some air in the piping of a refrigerating system, even if a very low vacuum is pumped when filling the system with the refrigerant. If the piping is simple and not very extensive, the circulation of the refrigerant will collect this air and carry it along with the vapor of the refrigerant into the compressor. After the air and vapor are compressed, they are carried along to the condenser, where the air separates from the refrigerant, for the reason that, as the latter condenses, it settles to the bottom of the condenser while the liberated air accumulates at the top. Usually, there is enough liquid refrigerant below the layer of air at the top of the condenser to prevent its passing out and circulating through the system; but, nevertheless, this air must be considered a hindrance to the operation of the plant, because that portion of the condenser occupied by the air cannot be reached by the vapor of the refrigerant. In other words, the effective condensing surface is reduced; and that surface which is still *available* is overworked to produce the proper condensing capacity. Another effect of this air accumulation is that more cooling water is used than is actually required for the normal operation of the condenser.

In large refrigerating systems with many feet of pipe in the condenser, if there is not enough liquid refrigerant in the condenser to seal its outlet, nearly all the air circulates and is compressed with the vapor of the refrigerant. This causes the cylinder and the discharge pipe of the compressor to become excessively heated, because, when air is compressed, its rise in temperature is greater than the increase in temperature when the refrigerants most commonly used are compressed. By thus increasing the temperature in the cylinder of the compressor, the presence of air brings about a corresponding increase in the discharge pressure. It must be remembered, too, that energy is lost and the efficiency of the system reduced by the continuous compression and expansion of such air which does no work in refrigeration.

A good method of ridding an ammonia refrigerating system of air is to attach a small pipe to the blowoff valve on the top of the condenser, so that an extension of this pipe is downward. In case an "equalizer" or some other pipe is also connected to this

blowoff valve, the equalizer may be shut off by closing its valve, so that the opening of the blowoff valve will not drain the refrigerant from other parts of the system. The end of the pipe which is attached to the blowoff valve should be immersed in a pail of cold water, with the open end a little below the surface. The blowoff valve should then be opened slightly, and large bubbles will form in the water and rise to the surface. These bubbles are due to the mixing of the air flowing from the condenser with a small amount of ammonia. The ammonia, however, does not rise but separates from the air and mixes with the water.[1] When all the air has been expelled, the flow through the pipe will be only ammonia vapor, which enters the water with a crackling sound and gives it a milky appearance. When the system has been cleared of air as nearly as possible, the blowoff valve should be tightly closed.

Non-condensable Gases in Condenser.—In purging condensers, consideration should be given to where the non-condensable gases should be expelled. It is a mistaken idea that all non-condensable gases accumulate at the top of the condenser. Since all systems using compressors require oil which decomposes at high temperatures, a certain amount of hydrogen gas will form and collect at the top of the condenser. However, there are other gases which are heavier and which will collect at the bottom of the condenser. These heavy gases eventually find their way into the liquid receiver and separate from the liquid refrigerant filling the upper part of the receiver, thus requiring the receiver to be purged.

The following table shows the densities for various gases found in condensers. The values are based on their weights at atmos-

Kind of Gas	Density
Hydrogen	1.00
Methane	7.96
Ammonia	8.50
Acetylene	13.00
Carbon monoxide	13.96
Nitrogen	14.01
Ethylene	14.10
Air	14.44
Oxygen	16.10
Carbon dioxide	21.95

[1] The ammonia and water in the pail may be set aside for cleaning purposes. It is a much stronger solution than ordinary household ammonia.

pheric pressure and 32° F. Other relationships can be expected within the condenser at working pressures. From the table it is seen that only a few gases often found in condensers are lighter than ammonia.

In the Voss double-pipe non-condensable gas separator, as shown in Fig. 183, the principle of sub-cooling the liquid ammonia is applied in such a manner that from the mixture of pure ammonia gas and non-condensable gases the ammonia is separated. This separation takes place in the *foul-gas jacket* of the separator by liquefying the pure ammonia at a pressure slightly below the condenser pressure, and at the intensely cold temperature of the

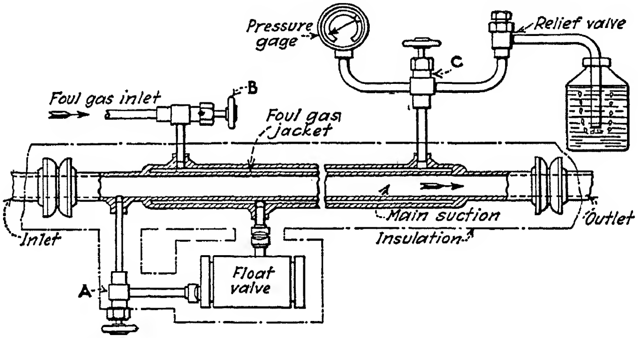

FIG. 183.—Voss double-pipe gas separator.

suction vapor going to the compressor through the inside pipe. The gas mixture in the jacket space contains a large percentage of foul gases. The separation of the mixture becomes complete when the foul gases have accumulated sufficiently to raise the pressure in the gas jacket high enough to open the automatic relief valve which permits the foul gases to escape into the atmosphere. This type of gas separator is designed to be a part of the main suction line of the compressor and operates in either a horizontal or a vertical position. The float valve, however, which allows the condensed ammonia to flow into the suction line must be placed horizontally. The relief valve is always located in a vertical position. The foul-gas connection leading into the valve B is preferably taken from the equalizing line on the condenser or may be taken from any point in the condensing system. Care should be taken, however, to make this connection at a reasonable distance from the hot discharge line coming from the compressor.

To start the gas separator, close tightly the adjusting nut on the relief valve by hand, and open the valves *A*, *B* and *C*. The pressure gage should then read the approximate condenser pressure. The valve *B* should next be closed to reduce the pressure to about 10 pounds per square inch gage lower than the condenser pressure; and the relief valve should be set to open at this pressure. The valve *B* should then be closed to reduce the pressure to about 25 or 30 pounds per square gage lower than the condenser pressure.

The mixture of ammonia vapor and non-condensable gases which enters the foul jacket are separated by the ammonia vapor condensing at a constant pressure corresponding to the low liquefaction temperature in the jacket. The liquid ammonia flows downward into the float chamber which causes the float to open the valve, allowing the liquid ammonia to escape into the suction line. The liquid ammonia in passing through the part of the gas separator in the suction line evaporates, thus cooling the foul-gas jacket.

This type of separator needs but little attention after it has been put in operation so that the usual hand-set expansion valve is not needed.

Defrosting Pipes.—Frost results from the moisture in the air's collecting and freezing on the cold pipes of a refrigerating system. If the coat of frost or ice is very thick on the pipes, the capacity of these pipes is reduced, the frost or ice acting as an insulator. If the frost is thin, as shown by a gray color of the pipes, their capacity is not much reduced. In case a heavy coat of frost occurs, it is necessary to lower the suction pressure in order to maintain the required temperature; and this puts more work on the compressor. Frost must be removed before it becomes too thick. In the direct system, this may be done by passing the vapor of the refrigerant *from the compressor* through the frosted coils for a short time. The frost or ice will then be loosened from the coils and may be scraped off. At times, it may be necessary to bypass the refrigerant from a section and let it stand idle until the frost is melted. When this method is to be used, additional coils should be operated while others are being freed from frost.

In laying out refrigerating coils, the pipes should not be placed close together, as the frost may extend from one pipe to the next, making the coil a mass of ice.

Sometimes, in the indirect system, the method is used of defrosting the cooling coils by allowing brine at ordinary temperatures to pass over their surfaces until the frost is melted. In order to do this, a spray pipe is placed over the coils; and from this spray pipe, the brine is sprayed on the top of each coil.

Amount of Piping Required.—When estimating the amount of piping to be used in refrigerating work, simple rules based on experience are generally used, for the reason that the data for the transmission of heat through the walls is too limited for accurate calculations. In general, a plant will operate more efficiently if the piping allowance is too large rather than barely enough. Because of this, manufacturers usually recommend the use of large amounts of piping. It should be remembered that large amounts of piping add to the first cost of the refrigerating plant.

Size of Pipe and Temperature Range.—If an insufficient amount of pipe is used in a refrigerating system, a greater cooling effect can be obtained by increasing the range of temperatures between the inside and the outside of the pipe. In the case of the direct system, this greater cooling effect can be obtained by lowering the suction pressure. On the other hand, a pipe having large surfaces can be used by decreasing the temperature range between the inside and outside of the pipe. In the direct system, this would be accomplished by increasing the suction pressure, thus reducing the work of the compressor. In general, then, it can be said that, in all cases of heat transfer, the smaller the range of temperature[1] the more efficient the heat transfer will be.

Size of Piping for Suction, Discharge, and Liquid Lines.—The laying out of headers and connections at the compressor should be given considerable care in order to eliminate changes of direction in the flow of the vapor or liquid, as changes of direction cause losses. The flow of vapor through cylinder ports as well as discharge and suction bends causes a loss of pressure. The area of a discharge port is sometimes designed for a velocity of about 10,000 feet per minute. In this case the ports should be straight, and large-radius bends should be used if there is a sudden change in direction.

[1] Economical working temperature differences are given in the table on page 256.

Size of Discharge Piping.—The size of piping in the discharge line of a compressor is determined by the quantity of refrigerant flowing in a given time. This can be calculated by finding the number of pounds of refrigerant circulated per minute and the specific volume of the refrigerant for the condition at the discharge port of the compressor. This quantity being in cubic feet per minute must be divided by the velocity in feet per minute in order to find the area of the pipe in square feet. The size of pipe required can then be found in Table XXV in the Appendix. The velocity of vapor through the discharge lines may be taken at about 6,000 to 7,000 feet per minute.

The size of discharge pipe is often computed by the use of the following equation based on the piston displacement

$$\text{Average velocity} = \frac{\text{piston area} \times \text{stroke} \times 2 \times \text{revolutions}}{\text{area of pipe}}$$

where the piston area is square inches, stroke is inches, revolutions are revolutions per minute, and area of the pipe in square inches. The above equation is for a double-acting or a twin-cylinder single-acting compressor.

Size of Suction Piping.—The design of the suction piping is more important than that of the discharge piping. In the discharge line the effect of pressure loss is to reduce the volumetric efficiency of the compressor and increase the work of compression; while wire-drawing in the suction line reduces the capacity of the compressor as well as increases the horsepower per ton of refrigeration, and reduces the volumetric efficiency. The design of the suction piping should be for velocities ranging from 2,000 to 5,000 feet per minute, depending on the length of the line.

Liquid Line.—The liquid line generally is not covered but in many cases it may be beneficial to do so. The advisability of having the liquid refrigerant as cool as possible has already been shown, and for this reason some plants have been equipped with special liquid coolers.

The size of the liquid line depends on its length. A velocity of about 3 to 6 feet per second can be used.

Pressure Drop in Ammonia Pipe Lines.—The drop in pressure in pounds per square inch $(P_1 - P_2)$ due to friction between the two ends of an ammonia pipe line may be found by the use of the following equation,

$$P_1 - P_2 = \frac{\quad}{144 \times 454 \times}$$

where V is the velocity in feet per *second*, L is the length of the pipe in feet, d is the diameter of the pipe in inches, and D is the density of the ammonia vapor in pounds per cubic foot.

Amount of Pipe for Cooling Coils.—In the indirect system of refrigeration, there are evaporator coils for cooling the brine and, also, brine coils for cooling the cold-storage compartments. If the brine is cooled in a tank, usually 120 to 150 running feet of 1¼ inch pipe are provided for each ton of refrigeration capacity. A double-pipe brine-cooling coil of the following dimensions has a rated refrigerating capacity of 15 tons every 24 hours: inner pipe, 2 inches in diameter; outer pipe, 3 inches in diameter; the pipes are 18 feet long, and the brine-cooling coils are 12 pipes high.

The refrigeration load of the cooling coils is the heat units (B.t.u.) to be removed from the walls of the building and from the goods stored. The amount of cooling surface may then be calculated by assuming that each square foot of cooling surface will pass about 3 B.t.u. per hour for each degree difference in temperature between the surfaces of the pipe.

The following tables have given ample refrigerating capacity in actual practice:

TABLE VI.—NUMBER OF CUBIC FEET OF WELL-INSULATED SPACE THAT CAN BE COOLED BY 1 RUNNING FOOT OF BRINE PIPE

For small rooms up to 1,000 cubic-feet capacity

Temperature held in rooms, degrees Fahrenheit	Diameter of pipe in coils			
	1 inch, cubic feet	1¼ inches, cubic feet	1½ inches, cubic feet	2 inches, cubic feet
0	½	⅝	¾	1
5	1	1¼	1½	2
10	2	2½	3	4
20	3	3¾	4½	6
32	4	5	6	8
36	5	6¼	7	10

TABLE VI.—NUMBER OF CUBIC FEET OF WELL-INSULATED SPACE THAT CAN BE COOLED BY 1 RUNNING FOOT OF BRINE PIPE (*Continued*)

For rooms from 1,000 to 10,000 cubic-feet capacity

Temperature held in rooms, degrees Fahrenheit	Diameter of pipe in coils			
	1 inch, cubic feet	1¼ inches, cubic feet	1½ inches, cubic feet	2 inches, cubic feet
0	1	1¼	1½	2
5	2	2½	3	4
10	3	3¾	4½	6
20	5	6¼	7½	10
32	7	8¾	10½	14
36	8	10	12	16

For rooms over 10,000 cubic-feet capacity

Temperature held in rooms, degrees Fahrenheit	Diameter of pipe in coils			
	1 inch, cubic feet	1¼ inches, cubic feet	1½ inches, cubic feet	2 inches, cubic feet
0	1½	2	2½	3
5	3	3¾	4½	6
10	4½	5⅝	6¾	9
20	6	7½	9	12
32	8	10	12	16
36	10	12½	15	20

TABLE VII.—NUMBER OF CUBIC FEET OF WELL-INSULATED SPACE THAT CAN BE COOLED BY 1 RUNNING FOOT OF DIRECT EXPANSION PIPE

For small rooms up to 1,000 cubic-feet capacity

Temperature held in rooms, degrees Fahrenheit	Diameter of pipe in coils			
	1 inch, cubic feet	1¼ inches, cubic feet	1½ inches, cubic feet	2 inches, cubic feet
0	½	⅝	¾	1
5	1	1¼	1½	2
10	2½	3⅛	3¾	5
20	4	5	6	8
32	6	7½	9	12
36	7	8⅜	10½	14

TABLE VII.—NUMBER OF CUBIC FEET OF WELL-INSULATED SPACE THAT CAN BE COOLED BY 1 RUNNING FOOT OF DIRECT EXPANSION PIPE (*Continued*)

For rooms from 1,000 to 10,000 cubic-feet capacity

Temperature held in rooms, degrees Fahrenheit	Diameter of pipe in coils			
	1 inch, cubic feet	1¼ inches, cubic feet	1½ inches, cubic feet	2 inches, cubic feet
0	1	1¼	1½	2
5	2	2½	3	4
10	4	5	6	8
20	6	7½	9	12
32	8	10	12	16
36	10	12½	15	20

For rooms over 10,000 cubic-feet capacity

Temperature held in rooms, degrees Fahrenheit	Diameter of pipe in coils			
	1 inch, cubic feet	1¼ inches, cubic feet	1½ inches, cubic feet	2 inches, cubic feet
0	1½	2	2½	3
5	3	3¾	4½	6
10	6	7½	9	12
20	10	12½	15	20
32	12	15	18	24
36	15	18¾	22½	30

In connection with the above tabulations, the following table shows the amount of cooling space that is provided in well-insulated refrigerator compartments per ton of refrigeration in 24 hours.

Temperature held in rooms, degrees Fahrenheit	Size of room		
	Up to 1,000 cubic feet, cubic feet per ton	From 1,000 to 10,000 cubic feet, cubic feet per ton	Over 10,000 cubic feet, cubic feet per ton
0	200	600	1,000
5	400	1,200	2,000
10	800	2,500	4,000
20	1,400	4,500	6,000
32	2,000	6,000	8,000
36	2,500	8,000	10,000

When the indirect system is used, a pump is needed to keep the brine in circulation. A centrifugal pump is adaptable for this service. It should be located near the expansion coils or brine cooler, and a bypass pipe should be connected to the discharge and suction ends of the pump. By regulatign a suitable valve in the bypass pipe, the flow of the brine may be regulated when the pump operates at a constant speed.

All of the pipes that make up the brine-cooling coils are generally connected to a header. The brine then flows through all of the pipes in the same direction. This method of connecting the pipes decreases the resistance to flow and reduces the work of the pump. The velocity of the brine should be about 60 feet per minute.

Length of Pipe for the Direct System.—In the direct system the size of the pipe will depend upon the conditions in the plant, such as the size of the rooms. Generally large pipes are used in large rooms while small pipes are used in small rooms. The length of the individual coils must be such as to allow the vapor to free itself from the pipe without too large a drop in pressure.

The maximum lengths for various sizes of pipe are shown in the following table:

Size of Pipe, Inches	Maximum Length for Direct System, Feet
3/4	900
1	1,100
1 1/4	1,300
1 1/2	1,500
2	1,900
2 1/2	2,300

Length of Pipe for Indirect System.—Brine coils are in general, arranged like the coils in the direct system. The length of the individual coils varies with the velocity of the circulating brine. For low temperature work a coil having 100 to 120 feet of pipe is fed by one regulating valve, while for high temperature work a coil having 400 to 440 feet of pipe is fed by one regulating valve. In general the size of pipe varies from 1 1/4 inches to 2 1/2 inches and the larger sizes are used for the larger rooms.

Heat Transfer Coefficients for Apparatus.—Table VIIA gives the heat transfer coefficients for various refrigerating apparatus in B.t.u. per square foot of surface per hour per degree Fahrenheit difference of temperature.

Salt Brines.—Water could be used in the cooling coils of the refrigerator in the indirect system if it were not for the fact that it freezes at 32° F. As it is often necessary to use cooling coil temperatures below this value, salt is dissolved in the water in order to lower the freezing point. Common salt (sodium chloride) or calcium chloride may be used. Calcium-chloride brine has the advantage that it has a lower freezing point than common-salt brine, and it will not corrode the pipes, as common salt will. In time, a brine of common salt will destroy the piping system. This corrosion can be reduced by adding sodium dichromate

TABLE VIIA.—HEAT TRANSFER COEFFICIENTS FOR REFRIGERATING APPARATUS

B.t.u. per square foot per hour per degree Fahrenheit

	B.t.u.
Can-ice-making piping:	
Old-style feed, nonflooded	15
Flooded	25
Ammonia condensers:	
Submerged (obsolete except for CO_2)	35
Atmospheric, gas entering at top	60
Atmospheric, drip or bleeder	125 to 200
Flooded	125 to 150
Shell and tube	150 to 300
Double pipe	150 to 250
Baudalot coolers (p. 476), counterflow, atmospheric type:	
Milk coolers	75
Cream coolers	60
Oil coolers	10
Water for direct expansion	60
Water for flooded	80
Brine coolers:	
Shell and tube	80 to 100
Double pipe	150 to 300
Cooling coils:	
Brine to unagitated air	2 to 2½
Direct expansion	1½ to 2
Water cooler:	
Shell and coil	30
Liquid ammonia cooler:	
Shell-and-coil accumulator	45 to 50
Air dehydrator:	
First coil, shell and coil (brine in coil)	5.0
Second coil, shell and coil (brine in coil)	3.0
Double pipe	6 to 7

to the common-salt solution. Because of the above-mentioned disadvantages in the use of common salt, calcium-chloride brine is the one more extensively used, even though it is the more expensive.

A strong solution of brine can be subjected to a lower temperature without freezing than can a weak solution. If the solution of brine is weak, it is likely to freeze to the cooling coil of the evaporator, thus reducing the heat transfer through the coils and, possibly, even stopping the circulation in the brine pipes. The brine solution should not be stronger than is necessary, since the specific heat of the brine is reduced as the density of the solution is increased. A very strong solution of brine requires a greater quantity to be circulated than does a weak solution, in order to absorb a given quantity of heat.

Different strengths of brine may be compared by their specific gravities. By *specific gravity* is meant the ratio of the weight of a given volume of brine to the weight of the same volume of water. For determining the specific gravities of brine a hydrometer called a *salinometer* marked with special calibrations for different strengths of brine is used. A salinometer is calibrated from 0 to 100°. The 0° corresponds to the specific gravity of water at 60° F., which is taken as the basis of measurement.[1] The 100° is taken as the specific gravity of a 25 per cent solution of salt brine at 60° F.

The properties of common-salt and calcium-chloride brine are shown in the following tables. From these tables one can also find the freezing points and the specific heats at 68° F. for different strengths of solutions.

Commercial calcium chloride contains about 20 per cent by weight of water, so that approximately 20 per cent more calcium chloride (by weight) than tabulated is required for solutions having the specific gravities given.

Saturated brine is a solution which cannot hold any more common salt or calcium chloride. If a solution is even nearly saturated, it is likely to deposit its salt in the pipes, thus interfering with the circulation and also insulating the pipes. Because of this, the usual "strength" of *common-salt* brine is 40 to 90° on the salinometer. Its lowest temperature is about 6° F. If

[1] This is an arbitrary scale. A more logical one would, for example, have 100° corresponding to a saturated solution of salt at 60° F., as no more salt could be held in solution at this temperature.

Table VIII.—Properties of Salt Brine*
Solution of sodium chloride in water
Specific heat of salt brine at the eutectic point (−4° F.) is 0.784

Per cent of salt (by weight)	Specific gravity†	Freezing point, degrees Fahrenheit	Specific heat for various percentages and temperatures, degrees Fahrenheit			
			+14	+32	+50	+68
5	1.036	26.7	.....	0.936	0.938	0.940
6	1.044	25.5	.....	0.924	0.927	0.929
7	1.051	24.2	.....	0.913	0.916	0.919
8	1.058	22.9	.....	0.902	0.906	0.909
9	1.066	21.6	.....	0.892	0.896	0.900
10	1.073	20.2	.....	0.882	0.887	0.890
11	1.081	18.8	.....	0.873	0.878	0.882
12	1.088	17.3	.....	0.865	0.869	0.873
13	1.096	15.7	.....	0.856	0.861	0.865
14	1.104	14.1	.....	0.848	0.853	0.857
15	1.112	12.4	0.835	0.841	0.846	0.849
16	1.119	10.6	0.827	0.834	0.839	0.842
17	1.127	8.7	0.821	0.827	0.832	0.835
18	1.135	6.7	0.815	0.821	0.825	0.828
19	1.143	4.6	0.809	0.815	0.819	0.822
20	1.151	2.4	0.804	0.809	0.813	0.815
21	1.159	0.0	0.798	0.803	0.807	0.809
22	1.167	− 2.5	0.794	0.798	0.801	0.803
23	1.176	− 5.2	0.789	0.793	0.796	0.798
24	1.184	+ 1.4	0.784	0.788	0.791	0.792
25	1.192	+13.3	0.779	0.783	0.786	0.787

* Specific heat, relative to water at 68° F., B.t.u. per pound per degree Fahrenheit.
† Specific gravity based on 60° F. water and 60° F. brine.

lower temperatures are desired, calcium-chloride brine should be used.

Corrosion Retarders.—In the construction of ice plants, all metals are subjected to corrosion which may be due to water, water solutions, oxygen, or air. Such corrosion is generally an oxidation of the metal surface by the direct action of oxygen in the air or by oxygen held in water. Electrolytic corrosion is the result of electric potential differences caused by the immersion of two different metals. This kind of corrosion may also result from stray electric currents. The corrosion resulting from the use of the so-called mixed brines (calcium and magnesium chlorides) is much greater than from brines made from either sodium chloride or calcium chloride. In new installations it is desirable

to have the brine in or near the neutral state which prevents excessive corrosion. A newly made calcium chloride brine always contains a small amount of alkali, which should be eliminated by mixing carbon-dioxide gas with the brine. In an ice tank, all that is necessary is to lead the carbon-dioxide gas into the brine near one of the agitators. The gas should be allowed to run very slowly into the brine at the bottom of the tank. The brine should be tested until there is no color present when one or two drops of phenolphthalein are added. The use of acids for neutralizing the alkalinity should be avoided as its use is likely to affect the ice-can coating. The carbon-dioxide treatment will cause a hard coating of carbonate of zinc to form on the galvanized surface of the ice cans which indicates the proper condition of the brine.

Calcium-chloride Brine Systems.—For calcium-chloride brine approximately 100 pounds of sodium dichromate ($Na_2Cr_2O_7 \cdot 2H_2O$) per 1,000 cubic feet of brine should be added (1.6 grams per liter) to retard corrosion. There should also be added sufficient sodium hydroxide to convert the dichromate to the neutral chromate. For neutral brines this amount is approximately 27 pounds per 100 pounds of dichromate; for ammoniacal brines the amount will be less and in some cases (when the brine shows red with phenolphthalein after the dichromate is added) none will be needed.

The sodium dichromate may be hung in a bag in the brine at a point of rapid circulation, or, when convenient, it is better to dissolve the dichromate retarder in a little warm water and pour the solution slowly into the brine. The bag method has the advantage of requiring less labor and insuring better distribution. Approximately one-half of the original amount of dichromate should be added to old brine once a year; analysis of the brine for dichromate would, of course, be a more advisable way to determine the quantity required, but the amount specified will usually be satisfactory.

In making up a new brine, it should be mixed in a clean tank, adding to the water and sodium dichromate (or chromate) sufficient caustic soda to make the brine just alkaline to phenolphthalein. The mixture should be allowed to stand until the insoluble portion settles. After this is done, the clear brine may be pumped into another clean tank, cooled to the usual temperature, and not until then should the cans be placed in the brine.

By following this procedure the abnormally high initial corrosion of the galvanizing on the cans can be largely eliminated.

Calcium-magnesium Brine Systems.—For calcium-magnesium chloride brines the same procedure recommended above should be followed, except that 200 pounds of dichromate per 1,000 cubic feet of brine should be used. The method of making up the new brine recommended above is especially advisable in this case.

Sodium-chloride Brine System.—Sodium dichromate is recommended for sodium brine systems in concentration of 200 pounds per 1,000 cubic feet of brine. The same procedure of adding alkali and renewing the dichromate annually is necessary.

Phosphate Treatment.—For sodium brines in open systems the use of disodium phosphate ($Na_2HPO_4 \cdot 12H_2O$) is suggested as an alternative. Although not so effective so dichromate and requiring frequent renewal, it does not have the disadvantage of irritating and poisoning the skin of workmen. The proper concentration to use is approximately 100 pounds per 1,000 cubic feet of brine, and this amount should be added once each month. The brine must be kept neutral or only slightly acid by the addition of muriatic acid if necessary. If phenolphthalein is used to test the brine and a pink color is observed, the acid should be added until the color just disappears. If the British Drug Universal indicator is used, the color should be yellow. The retarder must be dissolved in hot water and added slowly to the brine at a point in front of the agitator.

Fresh-water Recirculating Cooling Systems.—The use of sodium silicate (waterglass) is proper for this system. No universal rule can be given for the amount to use. In general, 1.5 gallons of 40° Bé. silicate per 1,000 cubic feet of make-up water should be used at first. This amount should cause the water to be pink when tested with phenolphthalein after 1 hour, and if this coloration is not obtained at the end of 1 week's application, the additions should be increased.

Sodium dichromate is also recommended for recirculating cooling systems, especially where the water comes in contact with corrosive industrial air. The amount required to stop corrosion varies considerably with the water and temperature and may be determined by immersing some clean steel in a continually aërated sample of the treated water for a week. With the proper additions, only very slight corrosion should be perceptible in that time. The recommended concentration for a first trial is

6 pounds per 1,000 cubic feet of make-up water. Caustic soda (1.75 pounds per 1,000 cubic feet) should also be added. Dichromate may be injurious to adjoining property when windage losses occur. It, therefore, has a narrower field of usefulness although it is more effective than the silicate. A large excess above the amount required to stop corrosion should be avoided.

Preparation of Brine.—A brine solution may be prepared in a barrel with a false bottom which is usually about 6 or 8 inches above the actual bottom. The false bottom is made up of strips of wood about 1 inch square in cross-section and placed about ½ inch apart. The strips are supported by two boards, 6 inches wide, placed edgewise, and nailed to the bottom. Over the false

TABLE IX.—PROPERTIES OF CALCIUM BRINE*
Solution of calcium chloride in water

Per cent of salt (by weight)	Specific gravity†	Freezing point, degrees Fahrenheit	Specific heat for various percentages and temperatures, degrees Fahrenheit −40	−22	−4	+14	+32	+50	+68
8	.069	24.2					882	0 887	0.892
9	.078	22.8					867	0 872	0.877
10	.087	21.4					853	0 858	0.863
11	.096	19.8					839	0 844	0.849
12	.105	18.2					825	0 831	0.836
13	.114	16.3					812	0 818	0.823
14	1.124	14.4					799	0 805	0.811
15	1.133	12.2				0.781	0.787	0 793	0.799
16	1.143	9.9				0.768	0.775	0.781	0.787
17	1.152	7.4				0.756	763	.770	0.775
18	1.162	4.7				0.745	752	.759	0.764
19	1.172	1.9				0.734	0.741	.748	0.754
20	1.182	- 1.0				0.723	0.731	0.738	0.744
21	1.192	− 4.0			0.704	0.713	0.721	0.728	0.733
22	1.202	− 7.3			0.695	0.704	0.711	0.718	0.724
23	1.212	−10.6			0.686	0 [illegible]	702	0.709	0.715
24	.223	−14.1			0.67[illegible]	[illegible]	[illegible]	[illegible].700	0.706
25	.233	−18.0		0.663	0.670	0.678	0.685	0.692	0.698
26	.244	−22.0		0.656	0.663	0.670	0.677	0.683	0.6[illegible]
27	.254	−27.0	0.643	0.649	0.656	0.663	0.669	0.676	0.682
28	.265	−32.0	0.636	0.642	0.649	0.656	0.662	0.	0.675
29	.276	−39.0	0.635	0.639	0.644	0.649	0.655	0 662	0.668
30	1.287	−46.0	0.631	0.635	0.638	0.643	0.648	0 655	0.661

* Specific heat, relative to water at 68° F., B.t.u. per pound per degree Fahrenheit.
† Specific gravity, based on 60° F. water and 60° F. brine.

bottom, burlap is placed and tacked to the sides of the barrel. This keeps the smaller particles of salt from dropping into the space below the false bottom. It also prevents any foreign matter contained in the water from getting into the brine. A 1¼-inch pipe, which is to serve as the inlet pipe for water, is connected to the barrel below the false bottom. The outlet pipe is placed about 6 inches below the top of the barrel and is about 1½ inches in diameter. It should be provided with a strainer. A piece of wire gauze placed over it will serve this purpose. The brine is made by filling the barrel with salt up to a point just below the outlet pipe. The water is then turned on and, in rising, dissolves some of the salt. The brine formed passes off through the overflow pipe. Gradually, the salt is dissolved, and more should be added to keep the barrel well filled.

A barrel like the one described can be connected to the brine system at its highest point. The strength of the brine can be varied by passing it through the barrel, adding salt to it to increase its strength, and adding water to weaken it. When the barrel is connected into the system, a bypass pipe with valves should be provided, so that the brine can be forced through or around the barrel. Considerable time is required to dissolve the common salt in making brine, and an even greater length of time is required with calcium chloride.

Calcium chloride comes fused in a solid mass, in sheet-iron drums containing about 640 pounds, and requires breaking up into lumps by hammering the outside of the drum before it is opened. It is also available and in a more convenient form as flakes.

Piping and Fittings for Refrigerating Systems.—The joints in an ammonia system are quite different from those ordinarily used for other purposes. The ordinary pipe joints cannot be made tight enough for most refrigerants. There are two forms suited to ammonia piping: (1) the gland joint and (2) the flange joint.

The *gland joint*, shown in Fig. 184, is simply a fitting which is threaded and has a recess filled with packing. A stuffing box is placed over the end of each pipe and is made tight against the packing by means of bolts. The end of the gland of the stuffing box which is next to the packing is beveled, so that the packing is forced against the end of the pipe, thus preventing leakage. If

the pipes are free from expansion and vibration, the packing may be a lead gasket; otherwise, a rubber gasket should be used.

The *flange joint* (Fig. 185) is made up of two flanges, one having a "tongue," and the other a groove, which fit together. Each

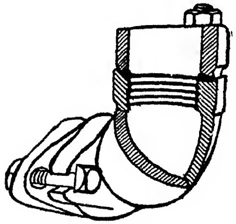

FIG. 184.—Flanged elbow for ammonia piping.

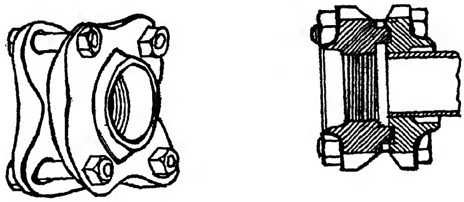

FIG. 185.—Flanges for pipe joints.

flange is threaded to receive the end of a pipe. A lead or rubber gasket is inserted in the groove. The gasket is compressed when

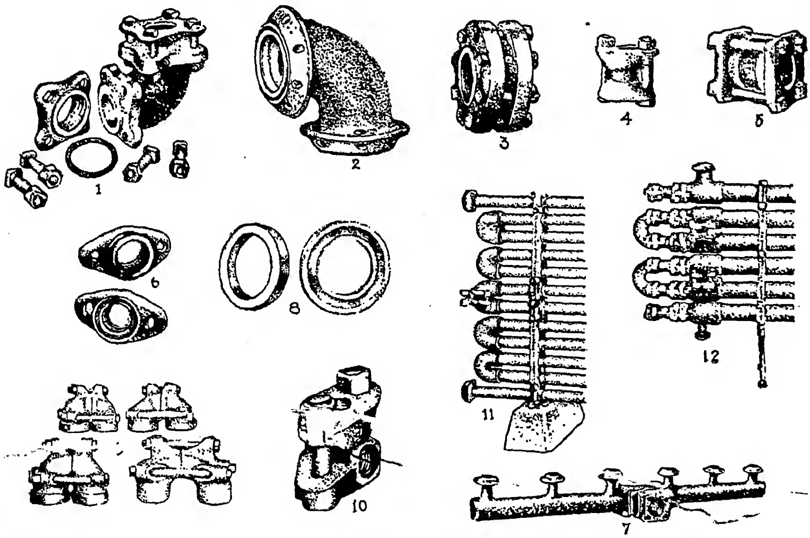

FIG. 186.—Typical pipe fittings for ammonia refrigerating system: (1) flanged elbow; (2) ground-joint bolted elbow; (3) bolted pipe flanges; (4 and 5) ground-joint union; (6) oval flanges; (7) branch tee or header; (8) joint rings; (9) split return bends; (10) oval flanged elbow; (11) solid and flanged return bends; (12) double-pipe connections for brine cooler.

the flanges are drawn together by bolts, thus preventing leakage. There is a recess in the backs of the flanges next to the pipe,

and this recess is filled with solder which prevents leakage along the pipe.

A *branch tee* or *manifold* of the kind used in refrigerating systems is marked (7) in Fig. 186. Other kinds of fittings are shown in the figure and explained by reference to numbers.

Working Temperatures in Ammonia Plants.—The temperature differences between the various elements of a refrigerating plant depend upon economic considerations. The cost of power, together with the cost of the pipe coil, determines, to a large extent, the magnitude of the temperature differences carried.

In general, it may be said that the larger the amount of coil surface the more economical the operating conditions will be, due to the fact that the suction pressure may be carried at a higher point. This is for the reason that the larger the coil surface for given conditions the smaller the temperature difference can be. The suction pressure should be carried as high as possible and still maintain the desired temperatures. The principal advantage of using a high suction pressure is that it requires less power per ton of refrigeration than do lower suction pressures. An additional advantage of higher suction is that the tonnage capacity of the compressor per cubic foot of displacement increases as the suction pressure is increased, since the ammonia weighs more per cubic foot.

In order to give an idea of the magnitude of these temperature differences, the following tables have been prepared, and these

TABLE IXa.—DIRECT EXPANSION, DEGREES FAHRENHEIT

Room temperature	−10	0	10	20	30	40	50	60
Ammonia temperature	−25	−15	−5	3	10	16	22	26
Temperature difference	15	15	15	17	20	24	28	34

BRINE SYSTEM,

Room temperature	−10	0	10	20			50
Brine temperature	−20	−12	−4	4			28
Temperature difference	10	12	14	16			22

Room temperature	−10	0	10	20	30	40	50
Ammonia temperature	−28	−20	−13	−6			13
Temperature difference	18	20	23	26	29	32	37

Brine temperature	−20	−12	−4	4		20	28
Ammonia temperature	−28	−20	−13	−6		8	13
Temperature difference	8			10		12	15

temperature differences should be maintained to insure economical operation.

Suction Pressure Required.—In order to operate a system at its maximum efficiency, the suction pressure should be as high as possible without failing to maintain the desired temperature. If different compartments are to be at different temperatures, the suction pressure must be of such value as to give the lowest temperature required. The evaporator of a refrigerator which is held at a higher temperature than other evaporators must be operated at a lower suction pressure than would otherwise be necessary.

CHAPTER VII

THERMODYNAMICS OF REFRIGERATING SYSTEMS

Refrigerating Machines Operating as "Heat Pumps."—A refrigerating machine is a mechanical device or "heat pump," which will transfer heat from a cold to a hotter body. This heat transfer, as stated by the second law of thermodynamics,[1] cannot take place of itself, but it can be effected by the expenditure of *mechanical work*. A steam, gas, or oil engine will serve as the heat pump of a refrigerating system if the engine is made to

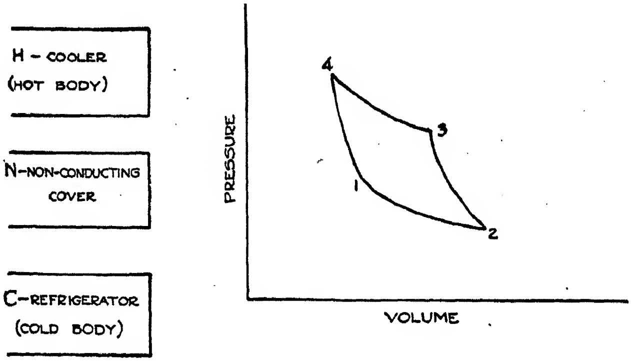

FIG. 187.—Diagram of Carnot cycle.

operate backward, so that the area of the indicator diagram taken on the cylinder of the engine represents work spent on, instead of done by, the gas or vapor which is used as the working substance.

A Carnot cycle in which *air is the working substance* may be used, as shown by the diagram in Fig. 187, to illustrate the "backward" operation of an engine, so that the cycle will be performed in the order indicated by the numbers 1-2-3-4.[2] Then, obviously, the area of the indicator diagram will have a negative

[1] For an explanation of thermodynamic principles, the reader is referred to Moyer, Calderwood, and Potter, "Elements of Engineering Thermodynamics," 4th Ed. This chapter uses the notation in that book.

[2] In the normal operation of a Carnot-cycle engine, the cycle is performed in the order 1-4-3-2, and then the area 1, 4, 3, 2 represents the work done by the air.

value and represents work spent upon the air. In the expansion 1-2, which is *isothermal,* meaning an expansion without change of temperature, the air is in contact with the *cold body C*, and it takes in a quantity of heat (Q_2) from the cold body equal to $wRT_2 \log_e r$,[1] where w is the weight of gas in pounds; T_2 is the constant absolute lowest temperature of the cycle in degrees Fahrenheit at which the expansion 1-2 takes place; T_3 is the constant absolute highest temperature in degrees Fahrenheit at which the isothermal compression 3-4 occurs; r is the ratio of expansion ($V_2 \div V_1$); and R is a constant depending for its value on the kind of gas. In the following compression 3-4, the air gives out to the hot body H a quantity of heat (Q_1) equal to $wRT_3 \log_e r$. There is no transfer of heat along the adiabatic lines 2-3 and 4-1. Thus, the cold body C is constantly being drawn upon for heat and can, therefore, be maintained at a lower temperature than its surroundings. At the lower temperature T_2, the amount of heat taken up by the air from the cold body C is $wRT_2 \log_e r$, and at the higher temperature T_3, the amount of heat given out by the air to the hot body H is $wRT_3 \log_e r$. In an actual *refrigerating machine operating with air,* the cold body C may consist of a coil of pipe through which brine circulates, and the cold refrigerated air is brought into contact with the outside of the coil. The brine may be kept, by the action of the refrigerating machine, at a temperature below 32° F., and this brine may be used to remove heat by conduction from water which is to be frozen to make ice. The hot body H or "cooler," which is only relatively hot with respect to the cold body C, is kept at a temperature as low as possible by circulating cool water around it. This circulating water absorbs the heat rejected to the hot body H by the "working" air in the system.

In the *reversed Carnot cycle* the term "efficiency" is in a sense a misnomer, as the object of a refrigerating system is to remove heat by expending energy. A better way to express this relation is by the ratio of the heat extracted to the work done which is called the *coefficient of performance.* The coefficient of performance for the reversed Carnot cycle may be represented by the equation,

$$\text{Coefficient of performance (C.P.)} = \frac{\text{heat removed } (Q_2)}{\text{work done } (Q_1 - Q_2)}$$

[1] 2.3 × log base 10 = log base e (Napierian or natural logarithms).

In another class of refrigerating machines, the working substance or refrigerant, instead of being air, is the vapor of a liquid, and the action proceeds by alternate evaporation at a low pressure of the liquid refrigerant to a vapor and then the condensation of this vapor at a high pressure. A refrigerant must be chosen which evaporates at the lower limit of temperature at a pressure not so low as to make the bulk of the compressor excessive.

The Air System of Refrigeration.—The dense- or closed-air system is illustrated in Fig. 188, in which air from which the

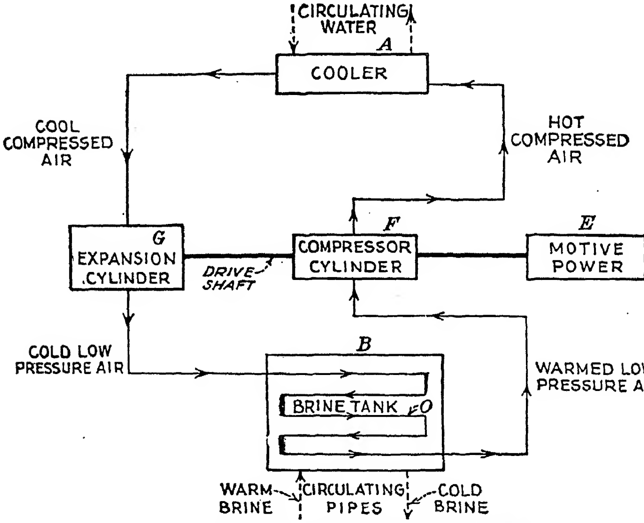

Fig. 188.—Outline of dense-air system of refrigeration.

moisture has previously been removed is continuously circulated. The engine E furnishes power[1] to drive the compressor F.

The cylinder of the compressor delivers hot compressed air into a cooler A, where it is cooled and then passes on to the expansion cylinder G (connected mechanically by a shaft to the compressor F and the engine E which supplies the motive power). From the expander, the cold low-pressure air passes on, first, through the cooling coils of the brine cooler in the tank B and then back to the compressor cylinder F; thus, the air cycle is

[1] Since the work done by the expansion of the cool compressed air is less than that necessary for compressing the air taken from the cooling coils of the evaporator for the same range of pressures, a means must be employed to make up for this difference; and for this purpose, a prime mover is used.

completed. The course of the circulating water and also of the brine is shown by the dotted lines with accompanying arrows.

The work performed in the cylinder of a compressor can best be studied by means of an *indicator diagram.* If the compression is performed very slowly in a cylinder which is a good conductor of heat, so that the air within may lose heat by conduction to the atmosphere as rapidly as heat is generated by compression, the compression is *isothermal,* meaning that it takes place at the constant temperature of the atmosphere. Now, if compressed air is distributed and used to do work in a compressed-air motor

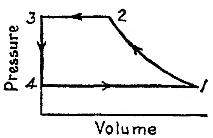

FIG. 189.—Compression diagram.

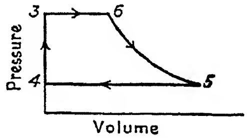

FIG. 190.—Expansion diagram.

or "expander" without change of temperature, and the process of expansion in the compressed-air motor or expander is also very slow and consequently isothermal, then (neglecting losses due to friction in pipes, etc.) there will be no waste of power in the whole process including compression of the air and its expansion. The indicator diagram would be the same per pound of air in the air compressor as in the compressed-air motor or expander, although, of course, the cycle of the compressed-air motor would be the reverse of that of the air compressor.

Adiabatic compression and expansion take place approximately if the compression and expansion are performed very quickly or when the air is not cooled during compression. In this case, the temperature of the air increases. The theoretical indicator diagram for adiabatic compression, as in Fig. 189, is 4-1-2-3, and that of the compressed-air motor or expander, in Fig. 190, is 3-6-5-4. The compression 1-2 and the expansion 6-5 (Figs. 189 and 190) are both adiabatic lines. As the result of the cooling of the compressed air between the compressor and the expander, the line 3-6 is shorter than the line 3-2.

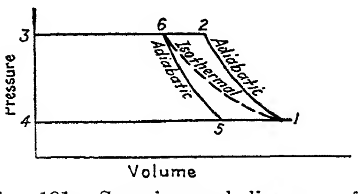

FIG. 191.—Superimposed diagrams of compressor and expander.

If the indicator diagrams of the compressor and the expander are superposed, as in Fig. 191, and then an imaginary isothermal line is drawn between the points 6 and 1, it will be easily seen that adiabatic compression causes waste of power, as indicated by the area 6-2-1, while adiabatic expansion in the compressed-air motor causes a further waste, as shown by the area 5-1-6.

Work of Compression.—Assuming no clearance in the compressor and that the compression is isothermal, the pressure-volume diagram of the compression is shown in Fig. 189; and the work W done in the cycle of compression in foot-pounds is represented by the following equations, where P_1 and P_2 are the absolute initial and final pressures in pounds per square foot, and V_1 and V_2 are the initial and final volumes in cubic feet:

$$W = P_1V_1 - P_1V_1 \log_e^* \frac{V_1}{V_2} - P_2V_2$$

which becomes, since $P_1V_1 = P_2V_2$,

$$W = -P_1V_1 \log_e \frac{V_1}{V_2} = P_1V_1 \log_e \frac{V_2}{V_1}$$

In practice, a compression cannot be made entirely isothermal. The difference between isothermal and adiabatic compression is

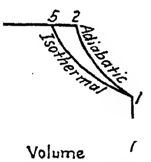

FIG. 192.—Diagram showing compression lines.

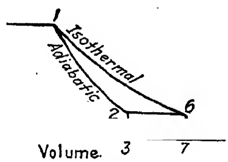

FIG. 193.—Diagram showing expansion lines.

shown graphically in Fig. 192 and between isothermal and adiabatic expansion in Fig. 193. In these examples, the terminal points are correctly placed for a certain definite ratio for both compression and expansion. In the compression diagram in Fig. 192, the area between the two curves 1-2-5 represents the work lost in the compression because of heating, and the area between

* 2.3 × log base 10 = log base e. Tables of natural logarithms are given in Moyer, Calderwood, and Potter, "Elements of Engineering Thermodynamics," 4th Ed., Appendix.

the two curves 1-6-7-3-2[1] (in Fig. 193) shows the work lost by cooling during the expansion. The isothermal curves are the same in the two cases.

Increase in the temperature of the air is, in a measure, prevented during the compression by cooling the cylinder of the compressor. This cooling of the cylinder has the effect of changing the compression curve. The curves, which would have been PV = a constant, if isothermal, and $PV^{1.4}$ = a constant, if adiabatic, will be very much modified. In *perfectly* adiabatic conditions, the exponent is 1.40 for air, but, in practice, the compressor cylinders are water jacketed, and thereby part of the heat of compression is conducted away, so that it becomes less than 1.40. This value varies with conditions and has generally a value between 1.2 and 1.3.

When the compression curve follows the law, PV^n equals a constant, and the work of compression(W) as in Fig. 189 is

$$W = \qquad n - 1$$

$$W = {}_{n} \quad (P_1V_1 - P_2V_2) \qquad \frac{n}{n-1} \times wR \mid T_1 -$$

where w is the weight of the gas or vapor in pounds.

The above formula when corrected for the friction loss may be written as follows:

$$rr \quad - \overline{n - 1}\, w$$

where f is the friction loss. Substituting the thermodynamic relation

$$K - 1^{*}$$

the formula becomes

$$W =$$

In the case of the expander the work done by the air in the cylinder can be obtained from this equation, providing the value

[1] The loss of work due to adiabatic expansion would be 1-6-7-3-2, if the isothermal and adiabatic expansions were interrupted at the points 6 and 2, respectively, and a further expansion were performed at constant volume to the level 4-7.

* The ratio of the specific heat of a gas at constant pressure (C_p) and the specific heat at constant volume (C_v) is K.

of n for the expansion line is known and the proper correction is made for the friction loss. The work done by the expander is then

W

It should be noted that the effect of friction here is to reduce the energy delivered to the shaft of the expander (acting as a motor).

If the value of n for the expansion line is 1.4 and the value of K is 1.4, the quantity $\left(\frac{K-1}{K}\right)\left(\frac{n}{n-1}\right)$ becomes unity.

The work done by the expander is then

The net work or the energy supplied from the driving unit is then equal to the difference of the work done by compressing the air and the work done during expansion. The net work supplied is then,

Net

Since the lines of compression and expansion are not isothermal, and therefore have values of n other than unity, it is often necessary to determine an unknown temperature. When two pressures and one temperature are known, the other temperature can be found by the use of the following equation:

where T_1 and P_1 are the initial temperature and pressure, P_2 the final pressure and n the exponential value for the line or path.

The Effect of Clearance upon Volumetric Efficiency.—It is impossible to construct a compressor without clearance; consequently, the indicator diagram of an operating compressor differs from the ideal. At the end of the discharge stroke, the clearance volume is filled with compressed vapor or gas of the refrigerant.

* The deviation of this equation may be found in Moyer, Calderwood, and Potter, "Elements of Engineering Thermodynamics," 4th Ed., p. 35.

When the piston moves on its outward stroke, the vapor or gas of the refrigerant expands, and the suction valves of the compressor will be closed until the piston has moved a sufficient distance to permit the trapped vapor or gas of the refrigerant to expand slightly below the suction pressure. When, in the expansion, the pressure reaches this value, any further movement of the piston opens the suction valves, and the vapor or gas of the refrigerant is drawn into the cylinder during the remainder of the stroke. Thus, the entire stroke of the compressor piston is not effective in pumping in a new supply of gas or vapor. The ratio of the apparent volume of vapor or gas drawn in, as shown

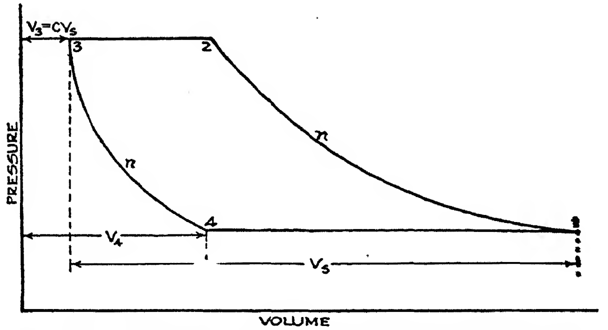

FIG. 194.—Diagram of ideal air compressor with clearance.

by the indicator diagram, to the volume swept by the piston, or piston displacement of the cylinder, is termed *apparent volumetric efficiency*. *True volumetric efficiency* is the ratio of the volume of gas or vapor actually drawn in to the piston displacement.

Figure 194 illustrates an ideal compressor diagram with clearance. The gas or vapor of the refrigerant which remains in the clearance space is

$$V_3 = CV_s,$$

where V_s = volume swept or piston displacement of the cylinder
C = percentage of clearance.

When the vapor or gas of the refrigerant expands to V_4, as shown in Fig. 194, the suction valves of the compressor are open and vapor or gas of the refrigerant is drawn into the cylinder as represented by the difference in volume between V_1 and V_4. This gas or vapor, as well as the clearance gas or vapor, is com-

pressed to point 2, while the compressed gas or vapor is discharged from points 2 to 3. Knowing the percentage of clearance, the volume swept by the piston (V_s), and the initial and final pressure, the volumetric efficiency due to clearance E_v may be determined from the following equations:

$$E_v = \frac{V_1 -}{V_s}$$

since

$$V_1 = V_s + CV_s,$$

$$V_1 - V_4 = V_s + CV_s - \left(\frac{P_3}{D}\right)^n CV_s$$

Therefore, volumetric efficiency is

$$E_v = \frac{- \left(\left(\frac{P_3}{D}\right)^n\right) | V_s}{V_s}$$

$$1 + C \Big| 1 - \left(\frac{P_3}{D}\right)^n$$

The volumetric efficiency E_v is less at low than at high suction pressures, because the weight of a cubic foot of vapor of the refrigerant decreases with the pressure.

The true volumetric efficiency may be expressed as the ratio of the capacity of the compressor to the piston displacement. The capacity is the actual amount of the vapor compressed and delivered, expressed in cubic feet per minute at intake temperature and pressure. The true volumetric efficiency is not easy to obtain as it requires the measurement of the refrigerant passing through the compressor, and takes into consideration the superheating of the suction vapor resulting from contact with the cylinder walls, piston and valves, which are always at a temperature above the suction vapor temperature. This effect cannot be shown by an indicator diagram. The superheating of the vapor causes a loss and, therefore, requires more work to be done

on the vapor and also reduces the capacity of the compressor. A compressor of the uniflow type (Fig. 73) is designed to reduce this loss.

The actual piston displacement can be determined if the volumetric efficiencies due to (1) clearance and (2) superheating effect are known. This can be expressed as

$$\textit{Actual Piston Displacement} = D_c \div (E_v \times E_s)$$

where E_v is the volumetric efficiency due to clearance, E_s the volumetric efficiency due to superheating, and D_c is the theoretical piston displacement.

The theoretical piston displacement per minute per ton of refrigeration D_{cmr} is expressed by the following equation

$$D_{cmr} = \frac{200V}{H_1 - h_3}$$

where H_1 is the total heat of vapor entering the compressor, h_3 the heat of the liquid of the refrigerant at the temperature it enters the expansion valve and V the specific volume of vapor for the conditions at the suction pressure.

The York Manufacturing Company made tests on slow-speed vertical single-acting compressors and found that the volumetric efficiency due to superheating could be expressed by the following empirical equation:

$$E_s = 1 - \frac{t_2 - t_1}{1{,}330}$$

where $t_2 - t_1$ is the rise of temperature during the compression stroke (see Fig. 197).

When dealing with vertical single-acting compressors in which the clearance has been made very small, the chief loss then becomes that due to superheating. For standard conditions, the volumetric efficiency of vertical single-acting compressors is about 85 per cent.

The *volumetric efficiency without clearance* was recently studied at the University of Illinois by Reed and Ambrosius. The results of their tests are shown in Fig. 195. These tests were made with a Worthington $4\frac{1}{2} \times 4\frac{1}{2}$-inch vertical single-acting high-speed compressor with feather (p. 116) valves. The cylinder was cooled by means of a water jacket. The measured piston displacement was 73.52 cubic inches while the clearance volume was 4.08 cubic inches or 5.56 per cent of the piston displacement. The tests

were run with varying suction pressures, holding the discharge pressure constant. The speed range for these tests varied from 515 to 540.

The piston displacement for the air system of refrigeration may be calculated for the compressor from the following equation,

$$D_c = \frac{wRT_1}{P_1\left[1 + C\left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right)\right]}$$

and for the expander D_e,

$$\left[1 + C\left(1 - \left(\frac{P_2}{P_4}\right)^{\frac{1}{n}}\right)\right]$$

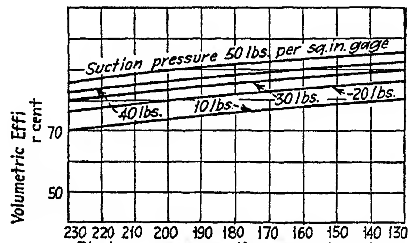

FIG. 195.—Volumetric efficiency without clearance for varying suction and discharge pressures.

where P_1 is the absolute suction pressure, P_2 is the pressure at the end of compression, w is the weight of air supplied per minute in pounds, C is the assumed ratio of the clearance to the piston displacement, R is the "gas constant" (53.3 for air), n is a thermodynamic exponent which is equal to 1.4 for adiabatic compression of air, T_1 is the temperature of the air at the beginning of compression, and T_5 is the temperature of the air at the end of the expansion (in the expander).

Problem.—A dense-air machine operates between the pressures of 65 pounds per square inch and 230 pounds per square inch absolute. The compressor receives air at a temperature of 10° F., and it is discharged from the *water cooler* at 95° F. The value for n for the compression line is 1.3; and for the expansion line, 1.4 (see Fig. 196).

Find (a) the weight of air per minute per ton of refrigeration; (b) net work per minute per ton of refrigeration; (c) weight of cooling water per minute per ton of refrigeration; (d) horsepower per ton of refrigeration; (e) displacement per minute per ton of refrigeration for the compressor, assuming 2 per

cent clearance; (*f*) displacement per minute per ton of refrigeration for the cooling coils or expander, assuming 2 per cent clearance; (*g*) coefficient of performace.

Assume the friction loss to be 15 per cent, and also assume the initial temperature of the cooling water to be 65° F. and the final temperature 75° F.

Solution:

$$\left.\frac{P_2}{P_1}\right)^{\frac{n-1}{n}},\ T_2 = 470\left(\frac{230}{65}\right)^{0.231} = 630° \text{ F. Abs. or } 170'$$

$$_4 = 555\left(\frac{65}{230}\right)^{0.286} = 387° \text{ F. Abs. or } -73° \text{ F.}$$

Weight of air per minute per ton of refrigeration

$$\frac{200}{0.24(10 - (-73))} \quad 10.0 \text{ pounds}$$

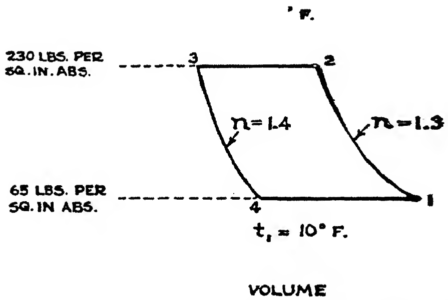

FIG. 196.—Graphical statement of problem.

(*b*) Net work per minute per ton of refrigeration

= 10 ×

$$- (1.00 - 0.15)(555 - 387)\Big]$$

= 216 B.t.u.

(*c*) Weight of cooling water per minute per ton of refrigeration

$$= \frac{0.24 \times 10(630 - 555)}{75 - 65} = 18 \text{ pounds}$$

(*d*) Horsepower per ton of refrigeration

$$= \frac{216 \times 778}{33{,}000} = 5.10$$

(*e*) Displacement per minute per ton of refrigeration for compressor

$$10 \times 53.34 \times 470$$

27.5 cubic feet

(*f*) Displacement per minute per ton of refrigeration for cooling coils or expander

$$10 \times 53.34 \times 387$$

$$= 22.7 \text{ cubic feet}$$

(*g*) Coefficient of performance

$$= \frac{10 \times 0.24(470 - 387)}{216} = 0.923$$

Related Physical Properties of Refrigerants.—There are five important properties applying particularly to vapors and gases that have important related characteristics. These are (1) pressure, (2) temperature, (3) volume, (4) total contained heat, and (5) entropy. These determine the characteristics of a vapor or gas of a refrigerant in any given state; and when any two of the five properties are known, the others may be obtained. Thus, if the pressure and temperature of a refrigerant are known, the three missing quantities (volume, total contained heat, and entropy) may be calculated or may be taken directly from suitable tables, as given in this book.

Example of Ammonia as Refrigerant.—The absolute pressure of ammonia vapor at a given condition is 170 pounds per square inch, and the temperature is 86.3° F. Now, if the temperature of this vapor is increased to 240° F. without changing the pressure, the other properties will be varied, as indicated in the following table:

	Case 1	Case 2
Pressure, pounds per square inch absolute	170	170
Temperature, degrees Fahrenheit	86.3	240
Specific volume, cubic feet per pound	1.76	2.47
Total contained heat, B.t.u. per pound	631.6	730.9
Entropy	1.19	1.351

The properties of saturated ammonia vapor are shown in Table I (pp. 492 to 495 inclusive in the Appendix). The properties of ammonia vapor are given with the temperature as the independent variable upon which the other properties depend. They have been determined by the U. S. Bureau of Standards.[1]

[1] *Bur. Standards, Bull.* 142, Table of Thermodynamic Properties of Ammonia.

In this table, the first column contains the even degrees of temperature in Fahrenheit. The second column contains the *absolute pressure* in pounds per square inch. And the third column contains the *gage pressure* in pounds per square inch.[1]

The pressures in the third column, which are below atmospheric, are given in inches of mercury below the standard atmospheric pressure (29.92 inches of mercury). It will be noted that the pressure increases gradually as the temperature is increased. The fourth column contains the volume of the saturated ammonia vapor in cubic feet per pound. The table shows that the specific volume increases rapidly as the temperature is lowered below 0° F. This fact is important in determining the size of the cylinder of a compressor, which is required to operate at very low temperatures. The fifth, sixth, and seventh columns show, respectively, the weight of the ammonia vapor in pounds per cubic foot, the total heat content of the liquid ammonia in B.t.u. per pound, and the total heat content of the ammonia vapor in B.t.u. per pound. These last two properties are useful for calculating the refrigerating effect of ammonia under different operating conditions. The latent heat of evaporation is shown by the eighth column and is given in B.t.u. per pound of ammonia. The latent heat of evaporation as shown in this last column represents the refrigerating effect which would be produced by the evaporation of 1 pound of liquid ammonia, provided that the liquid ammonia were initially at the saturation point. Under actual practical conditions, however, the temperature of liquid ammonia in a refrigerating system is invariably several degrees Fahrenheit above the saturation temperature, so that part of the latent heat of evaporation is unavoidably lost in cooling the remainder of the liquid ammonia to the temperature corresponding to the saturation point.

The ninth column gives the entropy of liquid ammonia in B.t.u. per pound per degree Fahrenheit absolute, and the tenth column gives the entropy of the ammonia vapor in the same units. It will be observed, in these last two columns, that the entropy of the liquid gradually increases as the temperature is increased, while the entropy of the ammonia *vapor* gradually decreases as the temperature is increased. It has already been explained that

[1] Gage pressure has been obtained from the absolute pressures in column 2 by subtracting a normal atmospheric pressure (14.7 pounds per square inch absolute) and dropping the last decimal place.

entropy is a mathematical ratio representing no physical condition of the substance and is used merely as a short-cut device in heat calculations.

Table II[1] is also a table of the properties of saturated ammonia, but, in this case, the absolute pressure in pounds per square inch is taken as the independent variable. The columns in this table are similar in Table I[1] with the exception that the ninth column has the heading Entropy of Evaporation. This column gives the entropy of evaporation expressed in B.t.u. per pound per degree Fahrenheit of *absolute temperature*.

Table III[1] gives the properties of saturated ammonia with the gage pressure as the independent variable. In all these tables, −40° F. has been adopted as a reference point for calculating the total heat contents. All heat contents above −40° F. are, therefore, in positive units. This use of a reference point eliminates minus quantities, which are sometimes awkward and lead to errors. In most of the calculations in refrigerating engineering, the temperatures are above −40° F., so that there is not likely to be occasion for the use of negative quantities for temperature conditions below −40° F.

Tables.—At the end of the book will be found tables and charts of the properties of ammonia and carbon dioxide, for certain temperatures and corresponding pressures, the latent heats of evaporation and the specific volumes in cubic feet per pound. These tables are for 1 pound of vapor, and the pressures are in absolute units; that is, the pressures are measured above a perfect vacuum. The ordinary gage indicates pressures above the atmospheric pressure. The absolute pressure, then, is the gage pressure plus the atmospheric pressure. The normal or average atmospheric pressure at sea level is 14.7 pounds per square inch. The values given in tables for latent heat are based on 1 pound of vapor; if more or less than a pound is used, these values must be multiplied by the actual weight of the vapor of the refrigerant in order to find the actual latent heat.

Entropy Table.—The theoretical condition of the vapor of a refrigerant during and after compression can be conveniently shown by means of entropy[2] calculations. The use of entropy

[1] Tables are given in the Appendix, pp. 496 to 503.

[2] *Entropy* is a mathematical ratio obtained by dividing the total amount of heat in 1 pound of a substance by the absolute temperature. Entropy will remain constant during an *adiabatic compression*, because, by definition,

for the calculation of refrigeration problems may be simply illustrated by the following example, which refers to an ammonia compression system. In this case, the ammonia vapor is assumed to be at the so-called *standard conditions;* that is the condensing temperature in the condenser is 86° F. and in the evaporator, 5° F. By reference to the tables on pages 492–512, the various properties of ammonia for the conditions of dry and saturated vapor before adiabatic compression and superheated vapor after adiabatic compression are given in the following items:

PROPERTIES OF AMMONIA VAPOR BEFORE COMPRESSION AT 5° F.

Pressure of saturated ammonia vapor, pounds per square inch absolute	34.2	
Specific volume per pound	8.15	cubic feet
Total heat of saturated vapor per pound	613.3	B.t.u.
Entropy of dry saturated vapor	1.3252	

PROPERTIES OF AMMONIA VAPOR AFTER COMPRESSION AT 86° F.

Pressure of saturated vapor, pounds per square inch absolute	170	
Specific volume per pound	2.346	cubic feet
Total heat of superheated vapor per pound	712.9	B.t.u.
Temperature of saturated vapor, degrees Fahrenheit	86	
Temperature of superheated vapor, degrees Fahrenheit	210	
Degrees of superheat of vapor, degrees Fahrenheit	124	
Entropy of superheated vapor	1.3252	

Action of Refrigerant in Evaporator.—It will be remembered that when a liquid evaporates, as in the cooling coils of the evaporator in a refrigerating system, it takes up heat. This heat is the latent heat of evaporation. Not all of this heat, however, is available for cooling purposes, because the temperature of the liquid on entering the expansion valve is at a higher temperature than that within the cooling coils of the evaporator. Some of the liquid must, therefore, be evaporated in order to lower its temperature. The evaporation necessary for this lowering of temperature of the refrigerant is a loss in the total available heat for refrigerating purposes and is explained more in detail on page 219.

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there is no heat transferred during a compression of this kind. The entropy of the gas or vapor of a refrigerant at the beginning of a compression stroke is the same as the entropy of the superheated gas or vapor at the end of an adiabatic compression.

When flowing through the expansion valve, the liquid ammonia at the higher pressure p_2 (Fig. 197) is converted into moist vapor at the pressure p_1 and quality x_4, with a reduction in temperature from the saturation temperature t_2 (corresponding to the pressure p_2) to the saturation temperature t_1 (corresponding to the pressure p_1). The moist vapor of the refrigerant absorbs heat from the substance which is being cooled, and all the liquid refrigerant carried in the vapor is evaporated. The amount of heat absorbed per pound of the refrigerant is $L_1(1 - x_4)$,[1] where x_4 is the quality of the ammonia vapor just after passing through the expansion valve and L_1 is the latent heat of evaporation at the pressure p_1.

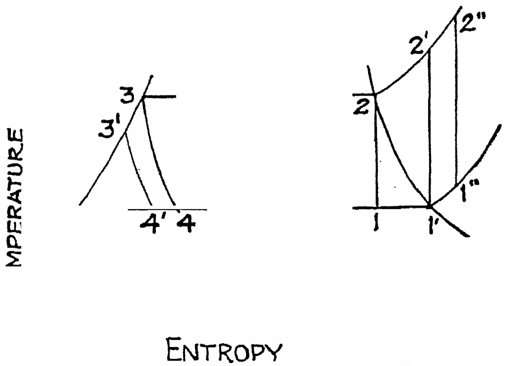


FIG. 197.—Entropy-temperature diagram showing effects of superheating and aftercooling.

Since the total heat of the liquid ammonia at the pressure p_2 equals the total heat of moist vapor of quality x_4 at the pressure p_1,

$$h_2 = h_1 + x_4L_1.$$

where h_2 is the heat of liquid at the pressure p_2, and h_1 is the heat of liquid at the pressure p_1. From this the initial quality x_4 just after the expansion valve is

$$h_2 - h_1$$

The theoretical pressure-volume (p.-v.) diagram for a refrigerating system, neglecting clearance and the effect of the expansion

[1] This statement applies only when the vapor leaving the evaporator is dry and saturated.

of the liquid in passing through the expansion valve, is shown by Fig. 198.

The vapor of the refrigerant is drawn into the compressor along the line 4-1 and is compressed adiabatically along the line 1-2. In compressing the vapor, the pressure and, also, the temperature increase. This puts the vapor into a suitable condition to be liquefied in the condenser by the cooling water. It is discharged from the compressor into the condenser along the line 2-3. After the liquid refrigerant passes through the expansion valve, as indicated by the line 3-4, it evaporates in the cooling coils of the evaporator, where it absorbs heat from the substances to be cooled. During this heat exchange, the refrigerant again becomes a vapor at the lower pressure p_1. At this pressure, there is,

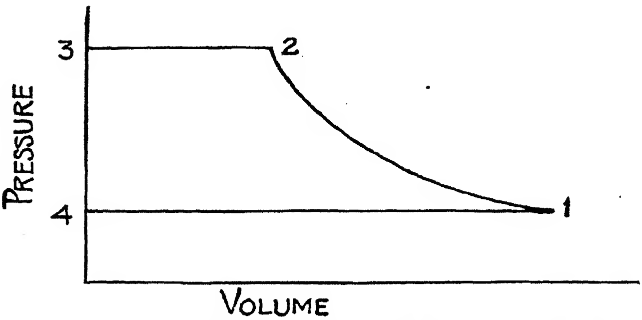


FIG. 198.—Typical pressure-volume diagram of compressor.

also, a correspondingly lower temperature at which the liquid refrigerant boils. This evaporation in the cooling coils of the evaporator causes a change in volume of the refrigerant, as shown by the line 4-1. This cycle appears in a temperature-entropy diagram, in Fig. 197.

The amount by weight of refrigerant that must be circulated in an ammonia refrigerating system per ton of refrigeration for the standard conditions of temperature (86 and 5° F., p. 106) can be calculated. The amount of heat removed from the substance in the refrigerator by the refrigerant as it evaporates in the cooling coils of the evaporator for these temperature conditions is (p. 287) 474 B.t.u. per pound of ammonia. One ton of refrigeration has a cooling effect at the rate of 200 B.t.u. per minute. The amount of refrigerant that must be circulated per

minute per ton of refrigeration at these standard conditions is, therefore, 200 ÷ 474 or 0.42 pound.

Horsepower Required for Compressor Cycle.—When the refrigerant has some liquid mixed with the vapor at the beginning of the compression stroke, the condition is called *wet compression*. On the indicator diagram, this kind of compression is shown by the line 1-2, in Fig. 197. When there is no liquid present; that is, when there is dry compression, the line showing compression is 1′-2′. On the other hand, when the vapor of the refrigerant is returned to the compressor with a small amount of superheat, the compression is 1″-2″, and the vapor is then discharged along 2′-3 or 2″-3 into the condenser, where it is changed to a liquid. The liquid refrigerant passes through the expansion valve, as indicated by the line 3-4, where the pressure and temperature are reduced. There is no loss or gain in heat along the 3-4 expansion line, so that this part of the refrigerating cycle is a constant-heat process. The refrigerant then takes up heat in the cooling coils of the evaporator when expanding along the line 4-1.

In some refrigerating machines, the cooling water of the condenser frequently cools the liquid refrigerant to a temperature lower than the temperature corresponding to the pressure. This *aftercooling* is shown in Fig. 197 by the lines 3-3′. It can be seen that this aftercooling of the liquid increases the available amount of refrigerating effect. A similar cycle can be shown on a *total heat-pressure* diagram (p. 285).

From the diagram in Fig. 198, it can be shown that the work done is the area 4-1-2-3. The following notation may be used to determine the work of adiabatic compression: I_1 and I_2 are the internal energies (B.t.u. per pound), respectively, of the refrigerant entering and leaving the compressor; v_1 and v_2 are the specific volumes of the vapor of the refrigerant (cubic feet per pound), neglecting, as being very small, the volumes occupied by the liquid particles of the refrigerant; p_1 and p_2 are the absolute suction and discharge pressures at the compressor (pounds per square foot); A is the heat equivalent of mechanical energy or $1/778$ (B.t.u. per foot pound). Then, as shown in Fig. 198, the work performed on the refrigerant under the line 1-2 is $I_2 - I_1$; the work performed under the line 2-3 is Ap_2v_2; the work done by the refrigerant under the line 4-1 is Ap_1v_1. The *net work* of compression is therefore, $I_2 - I_1 + Ap_2v_2 - Ap_1v_1 = (I_2 + Ap_2v_2) - (I_1 + Ap_1v_1)$. But, since $I_2 + Ap_2v_2 = H_2$, the total

heat (B.t.u. per pound) of the refrigerant at 2 and $I_1 + Ap_1v_1 = H_1$, the total heat (B.t.u. per pound) at 1, it follows that the work of compression is simply the difference between the total heats of the refrigerant at the points 2 and 1, or

Work of compression per cycle $= H_2 - H_1$ (B.t.u. per pound)

The operation of the compressor is, theoretically, a reversed Rankine cycle.

The vapor discharged by the compressor may be *superheated*, having a temperature t_s at a pressure P_2. The condition of the discharged vapor is determined by equating the entropies at the inlet and discharge pressures. The total heat (H_2) of the discharged vapor will be

$$H_2 = [h_2 + L_2 + C_p(t_s - t_2)]$$

where C_p is the specific heat at constant pressure.

The *theoretical* horsepower of the compressor is, if w pounds of vapor of the refrigerant are circulated per minute;

$$\text{Hp.} = \frac{w(H_2 - H_1)778}{33{,}000} = \frac{w(H_2 - H_1)^*}{42.42}$$

Theoretical Horsepower Required for Adiabatic Compression of Ammonia.—In the table (p. 273) the total amount of heat in the ammonia vapor before compression is 613.3 B.t.u. per pound. After adiabatic compression, the total heat in the superheated ammonia vapor is 712.9 B.t.u. per pound. There is, therefore, an increase of 99.6 B.t.u. per pound of ammonia during the adiabatic compression.

There is a definite mechanical equivalent for every heat unit expended in compression or any other kind of work, the heat equivalent of 1 horsepower being 42.42 B.t.u. per minute.[1]

The expenditure of 99.6 B.t.u. per pound of circulated ammonia vapor is equivalent, therefore, to 99.6 ÷ 42.42 or 2.35 horse-

* In the case of fluids of which there are no published tables and charts, the theoretical horsepower can be calculated (if the compression is assumed to be adiabatic) from the following formula:

$$\text{Hp.} = \frac{144n}{33{,}000(n - 1)} \times P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

where V_1 = volume compressed per minute, cubic feet; P_1 = absolute intake pressure, pounds per square inch; P_2 = absolute discharge pressure, pounds per square inch; and n = exponent of PV^n = constant.

[1] One horsepower is defined as 33,000 foot-pounds per minute; similarly, 1 B.t.u. has the mechanical equivalent of 778 foot-pounds. One horsepower, therefore, in heat units is 33,000 ÷ 778 or 42.42 B.t.u. per minute

power per pound of ammonia. In the refrigerating system which is the basis of the example on page 276, it is found that 0.42 pound of ammonia is required to be circulated per minute per ton of refrigeration, and, therefore, the horsepower required per unit of refrigeration is 2.35 times 0.42 or 0.99 horsepower per ton of refrigeration. In the calculation of this *theoretical horsepower*, it is assumed that the cylinder of the compressor is filled completely at each suction stroke of the piston.

Theoretical Horsepower Required for Compressors When Corrected for Volumetric Efficiency.—The horsepower required to drive a compressor per ton of refrigeration may be approximately calculated by correcting the theoretical horsepower requirement, as determined in the last paragraph for the volumetric efficiency, which, in the case of a *vertical single-acting compressor* with practically no clearance, may be assumed to be 84 per cent. The theoretical power requirement in the preceding example, for an ammonia compressor, as thus corrected, is $0.99 \div 0.84$ or nearly 1.2 horsepower per ton of refrigeration.

On the other hand, if the compressor is *double-acting* or is a vertical type with about the same amount of clearance provided in a *horizontal compressor*, the volumetric efficiency would be about 80 per cent, and the actual indicated horsepower would probably be about $0.99 \div 0.80$ or nearly 1.25 horsepower per ton of refrigeration.

Actual Horsepower Required to Drive Compressor.—There are various losses in the compressor, such as friction, windage, etc., that increase the power actually required to drive a compressor. These losses are about 20 per cent of the actual horsepower. In the two cases above, the actual horsepower required to drive the vertical single-acting compressor is $1.2 \div 0.80$ or 1.5 horsepower per ton of refrigeration, and in the case of the double-acting compressor, it is $1.25 \div 0.80$ or 1.6 horsepower per ton of refrigeration.

Kilowatts Required to Drive Compressor.—In cases where compressors are to be driven by electric motors, it is necessary to compute the electric power required. This transposition of power units can be made by multiplying actual horsepower by 0.746. This may be done in the two cases of the ammonia compressors under discussion, (1) for the vertical single-acting compressor with small clearance, for which the electric power required is 1.5×0.746 or 1.12 kilowatts per ton of refrigeration; and (2)

in the case of the double-acting compressor with normal clearance, for which the electric power required is 1.6×0.746 or 1.19 kilowatts per ton of refrigeration.

Heat Absorbed by Vapor in Evaporator.—The heat absorbed per minute by w pounds of vapor passing through the cooling coils of the evaporator will be

$$Q_r = w(1 - \qquad \text{(B.t.u. per pound)}$$

when the refrigerant leaves the evaporator as a dry and saturated vapor, or is expressed by

$$Q_r = w[(1 - \qquad + C_p(t_1'' \qquad \text{(B.t.u. per pound)}$$

when the compression is "dry"; that is, the refrigerant leaving the evaporator is *superheated* to the temperature t_1''. For the case of "*wet*" compression, that is when the refrigerant entering the compressor is a moist vapor of quality x_1, the heat absorbed in the evaporation is given by

$$Q_r = \qquad - x_4)L_1$$

and in general,

$$Q_r = w(H_1 - h_4) \text{ (B.t.u. per pound)}$$

Heat Absorbed by Cooling Water in Condenser.—The heat given to the cooling water in the condenser is, in general, as indicated in Fig. 197.

$$Q_c = w(H_2 - h_3) \text{ (B.t.u. per pound)}$$

where h_3 is the heat of the liquid at point 3 (temperature t_2). If the vapor leaving the compressor is superheated to the temperature t_2;

$$Q_c = w[C_p(t_2'' - t_2) + L_2 + (h_3 - \qquad \text{(B.t.u. per pound).}$$

In this equation, L_2 is the latent heat of evaporation at the pressure p_2 and h_3' is the heat of the liquid due to aftercooling in condenser.

Theoretical Coefficient of Performance.—The theoretical coefficient of performance is the ratio of the heat absorbed by the refrigerant in the evaporator to the heat equivalent of the work done by the compressor. The theoretical amount of heat resulting from adiabatic compression per ton of refrigeration per minute is 99.6×0.42 B.t.u. The heat equivalent of 1 ton of

refrigeration is 200 B.t.u. per minute. The coefficient of performance is, therefore, $200 \div (99.6 \times 0.42)$ or 4.78.

Problem.—An ammonia compressor operates with dry compression under standard conditions. If the vapor is superheated to a *temperature of* 20° F. when it enters the compressor, determine the following, assuming the liquid ammonia at the throttling valve to be 80° F. with a 10° F. rise in temperature of the cooling water. Also, assume an overall efficiency of 75 per cent and a volumetric efficiency of 80 per cent:

Find (*a*) weight of ammonia per minute per ton of refrigeration; (*b*) horsepower required by compressor per ton of refrigeration; (*c*) gallons of cooling water per minute per ton of refrigeration; (*d*) piston displacement per minute per ton of refrigeration; (*e*) coefficient of performance.

Solution:

(*a*) Weight of ammonia per minute per ton of refrigeration

$$= \frac{200}{622.2 - 132}$$
$$= 0.408$$

(*b*) Horsepower per ton of refrigeration

$$= \frac{0.408[726 - 622]778}{33{,}000 \times 0.75 \times 0.80}$$
$$= 1.66$$

(*c*) Gallons of cooling water per minute per ton of refrigeration

$$= \frac{0.408[726 - 132]}{8.33 \times 10}$$
$$= 2.91$$

(*d*) Piston displacement per minute per ton of refrigeration

$$= \frac{0.408 \times 8.473}{0.80}$$
$$= 4.32 \text{ cubic feet or } 7{,}465 \text{ cubic inches.}$$

(*e*) Coefficient of performance

$$= \frac{200}{0.408[726 - 622] \div [0.75 \times 0.80]}$$
$$= 2.83$$

Mean Temperature Difference.—When heat flows from a hot fluid to a cool fluid and the temperature of each fluid is changing, it is to be noted that the actual mean temperature difference is a progressive average between the changing temperatures. This progressive average is not the arithmetical mean, and therefore it must be determined by higher mathematics.

In the case of an *ammonia condenser*, the cooler fluid (water) has its temperature increased, and the hotter fluid (ammonia) has its temperature decreased. This is true in a counter- or parallel-

flow condenser. In a brine cooler the temperature of the colder fluid is constant while the temperature of the hotter fluid is varying. In such and similar apparatus the *mean temperature difference*, D_m, may be calculated by the following equation:[1]

$$D_m = \frac{D_a - D_b}{\log_e \frac{D_a}{D_b}} = \frac{D_a - D_b}{2.3 \log_{10} \frac{D_a}{D_b}} = \frac{(T_a - t_a) - (T_b - t_b)}{2.3 \log_{10} \frac{T_a - t_a}{T_b - t_b}}$$

where $D_a = T_a - t_a$ is the temperature difference at end a of the surface, and $D_b = T_b - t_b$ is the temperature difference at end b of the surface, as shown in Fig. 199.

The mean-temperature-difference equation stated above is sometimes called the *logarithmic mean-difference equation* and is often written in a form of equation which is based upon a constant liquefaction temperature throughout the condenser and also a uniform heat transfer throughout the condenser.

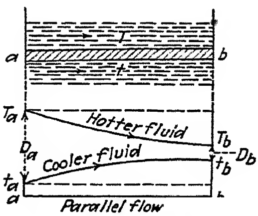


FIG. 199.—Diagram to illustrate mean temperature difference.

The surface of a condenser through which the heat is transmitted to the cooling water may be divided into the superheat, liquefaction, and aftercooling sections. In each of these sections the heat-transfer coefficients and the mean temperature difference must be known, so as to determine the required amount of surface for each section and the total heating surface.

The sectional heat-transfer coefficients of a condenser are not often known, so that another method is generally used to compute the surface of the condenser. This method requires the determination of the heat removed in the superheater, liquefier and aftercooler sections. It also is necessary to find the mean temperature differences for each section.

The average mean temperature difference (t_d) in the whole condenser may be calculated by the following equation:

$$t_d = \frac{H_a + H_L + H_s}{\frac{H_a}{t_a} + \frac{H_L}{t_L} + \frac{H_s}{t_s}}$$

[1] For the derivation of the mean-temperature-difference equation see Hirshfeld and Barnard, "Elements of Heat-Power Engineering," 2d Ed., pp. 636–650.

where H_c, the heat removed in the condenser, is $H_a + H_L + H_s$; H_a, H_L and H_s are the heats removed in the aftercooler, liquefier and superheater sections, respectively; and t_a, t_L and t_s, are the mean temperature differences in the aftercooler, liquefier, and superheater sections, respectively.

Heat Removed in Different Sections of Condenser.—It has previously been shown that the total heat removed in the condenser is made up of the heat of superheat, latent heat, and heat removed from the liquid to aftercool it. Because of these divisions the condenser may be thought as divided into the superheater, liquefier, and aftercooler sections.

Heat removed in liquefaction section per minute per ton of refrigeration:

$$Q_L = \frac{200 \times L_2}{Q_r}.$$

Heat removed in aftercooler section per minute per ton of refrigeration:

$$Q = \frac{200(h_3 - h_3')}{Q_r}.$$

Heat removed in superheater section per minute per ton of refrigeration:

Calculation for Condenser.—Problem: An ammonia double-pipe condenser is supplied with ammonia at a pressure of 150 pounds per square inch absolute and at a temperature of 200° F. If the suction pressure is 32 pounds per square inch absolute and the temperature at the expansion valve is 75° F., find the following when the cooling water enters at a temperature of 60° F. and leaves at 70° F. Assume the mean coefficient of heat transfer to be 150 B.t.u. per square foot per hour per degree Fahrenheit. Find the following:

a. Heat removed in liquefaction section per minute per ton of refrigeration.

b. Heat removed in aftercooler section per minute per ton of refrigeration.

c. Heat removed in superheater section per minute per ton of refrigeration.

d. Mean difference of temperature for each section.

e. Average mean difference of temperature.

f. Area in square feet per ton of refrigeration.

Solution:

a. Pounds of ammonia per minute per ton of refrigeration

$$\frac{200}{612.4 - 126.2} \quad 0.41 \text{ pound.}$$

Heat removed in liquefier section per minute per ton of refrigeration

$$= 0.41 \times 499.9 = 205 \text{ B.t.u.}$$

b. Heat removed in aftercooler section per minute per ton of refrigeration

$$= 0.41(130.6 - 126.2) = 1.81 \text{ B.t.u.}$$

c. Heat removed in condenser per minute per ton of refrigeration

$$= 0.41(708.9 - 126.2) = 239 \text{ B.t.u.}$$

Heat removed in superheat section per minute per ton of refrigeration

$$= 239 - 205 - 1.81 = 32.19 \text{ B.t.u.}$$

d. The temperature of the water at the end of the aftercooler and superheater sections may be found as follows:
Temperature of water leaving aftercooler section

$$= 60 + \frac{1.81}{239}(70 - 60) = 60.076^\circ \text{ F.}$$

Temperature of water leaving the liquefaction section

$$0 - 60) = 68.65^\circ \text{ F.}$$

Mean temperature of aftercooler section

$$= \frac{(78.81 - 60.076) - (75 - 60)}{2.3 \log_{10}\left(\frac{78.81 - 60.076}{75 - 60}\right)}$$
$$= 13.9^\circ \text{ F.}$$

Mean temperature in liquefaction section

$$= \frac{(78.81 - 60.076) - (78.81 - 68.65)}{2.3 \log_{10}\left(\frac{78.81 - 60.07}{78.81 - 68.65}\right)}$$
$$= 14.01^\circ \text{ F.}$$

Mean temperature in superheat section

$$= \frac{(200 - 70) - (78.81 - 68.65)}{2.3 \log_{10}\left(\frac{200 - 70}{78.81 - 68.65}\right)}$$
$$= 47^\circ \text{ F.}$$

e. Average mean temperature difference

$$t_d \quad \frac{239}{\frac{1.81}{13.9} \quad \frac{205}{14.01} + \frac{32.19}{47}}$$
$$= 15.5^\circ \text{ F.}$$

f. Area in square feet per ton of refrigeration

$$= \frac{239 \times 60}{150 \times 15.5}$$
$$= 6.15 \text{ square feet.}$$

Heat Balance of the Compression System.—The general formula for the heat balance is as follows:

$$H_e + . \qquad . + H_3$$

where H_e = heat absorbed by evaporating refrigerant; H_w = heat equivalent of work in compressor; H_1 = heat rejected in condenser; H_3 = heat rejected or radiated in addition to H_1.

For purposes of illustration, the following list of quantities involved in the computation of the heat balances of compound compression systems is given:

Heat Absorbed (B.t.u. per Hour)

a. Heat absorbed in evaporator.
b. Heat entering evaporator insulation.
c. Heat absorbed in low-pressure suction main.
d. Heat absorbed in low-pressure suction trap.
e. Heat equivalent of work done in compressor.
f. Heat absorbed from engine room through cold surface of low-pressure compressor.
g. Heat absorbed through surface of intermediate liquid receiver.
h. Heat absorbed through surface of intermediate-pressure liquid line.
i. Heat absorbed from engine room through surface of high-pressure suction main.
j. Heat absorbed through cold surfaces of high-pressure compressor.
k. Heat absorbed or rejected through condenser shells.
l. Heat absorbed or rejected through receivers.
m. Heat absorbed or rejected in high-pressure liquid line.

Heat Rejected (B.t.u. per Hour)

n. Heat rejected by hot surface of low-pressure compressor.
o. Heat rejected from low-pressure discharge main between low-pressure compressor and intermediate vapor cooler.
p. Heat rejected in intermediate vapor cooler.
q. Heat rejected to engine room by intermediate vapor cooler.
r. Heat rejected in discharge main from intermediate vapor cooler to intermediate liquid receiver.
s. Heat rejected by hot surfaces of high-pressure compressor.

t. Heat rejected by high-pressure discharge main and oil separator between machine and condensers.

u. Heat rejected in ammonia condensers.

v. Heat rejected in liquid cooler.

w. Heat rejected or absorbed through condenser shells.

x. Heat rejected or absorbed through receivers.

y. Heat rejected or absorbed in high-pressure liquid line.

The heat absorbed by the condenser is theoretically equal to the sum of the heat absorbed in the refrigerator and the heat equivalent of the work of compression.

The volume of the ammonia vapor delivered to the compressor can be readily found from the weight of the refrigerant which circulates in the system and the specific volume of the vapor when it enters the compressor. The actual displacement of the compressor can then be ascertained if the volumetric efficiency is known.

Pressure-total Heat Chart of Ammonia.—A useful chart for refrigerating calculations when ammonia is used has been prepared by the U. S. Bureau of Standards. In it, the absolute pressure of the ammonia is taken as the scale of ordinates, and the total heat in B.t.u. per pound is the scale of abscissas. A chart of this kind is shown in Fig. 292 (Appendix). A simplified diagram is shown in Fig. 200. For the diagram enclosed by heavy lines, the temperatures of saturated ammonia are taken at the so-called *standard* values of 86 and 5° F., respectively, in the condenser and the cooling coils of the evaporator. The intersections of the constant-temperature lines in the diagram with the *saturated-liquid* line (shown by the very heavy line on the chart) show on the scale of ordinates the *absolute* pressure in pounds per square inch corresponding to the temperature of the saturated condition of the ammonia vapor. Using this method of determining pressures, it will be found by interpolation that the constant-temperature line for the standard condition of 86° F. intersects the saturated-liquid line at approximately 169 pounds per square inch absolute pressure; and, similarly, that the 5° F. constant-temperature line intersects the saturated-liquid line at 34 pounds per square inch absolute pressure. The point corresponding to 86° F. on the saturated-liquid line is marked *A* in the figure. The first process to be represented in the figure is, of course, the expansion through the expansion valve from the higher to the lower pressure, that is, from 169 pounds per square inch

absolute to 34 pounds per square inch absolute. Since the expansion is without any change in the total amount of heat per pound of ammonia, a vertical line on the chart will represent the expansion. This vertical line is, of course, drawn through the point marked A to intersect the pressure line corresponding to the pressure in the cooling coils of the evaporator, that is, 34 pounds per square inch absolute. The intersection of the vertical line through A with the pressure line corresponding to 34 pounds per square inch absolute is marked B.[1] The diagram also shows curved lines marked "constant quality."[2] The

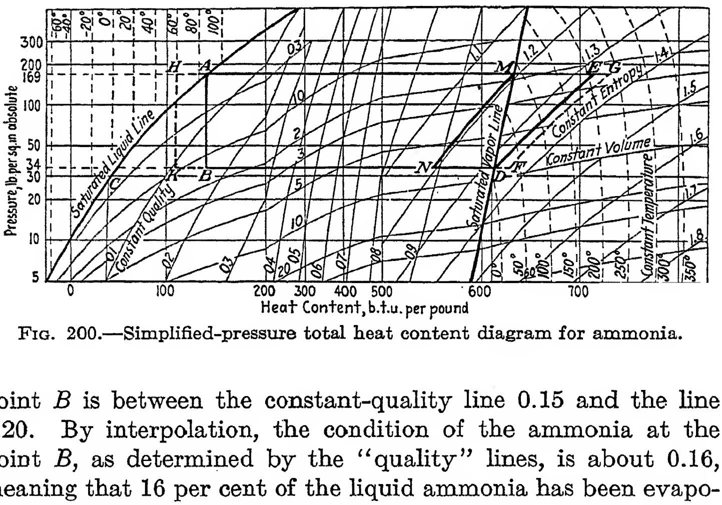


FIG. 200.—Simplified-pressure total heat content diagram for ammonia.

point B is between the constant-quality line 0.15 and the line 0.20. By interpolation, the condition of the ammonia at the point B, as determined by the "quality" lines, is about 0.16, meaning that 16 per cent of the liquid ammonia has been evaporated when it passes through the expansion valve. In the cooling coils of the evaporator, the mixture of liquid ammonia and ammonia vapor absorbs heat from the substance being cooled in the refrigerator and, in thus taking up or absorbing heat, becomes

[1] It will be noticed that the point B is on a horizontal line through the point where the 5° F. constant-temperature line intersects the saturated liquid line is marked C.

[2] If the horizontal line through the point B is extended to the point D on the saturated-vapor line, the condition of the ammonia as represented by the point D will be shown by the constant-quality lines to be 1.00 meaning that at this point, the ammonia is 100 per cent vaporized. It is interesting to note that the quality of the ammonia at any point in a chart of this kind can be determined as at B, by measuring the distances CD and CB and then calculating the ratio, or $CB \div CD$ is the quality of the ammonia.

completely vaporized, so that when it passes out of the cooling coils or evaporator, its condition is represented on the chart in Fig. 200 by the point *D*. It is interesting to calculate now from the chart how much heat has been lost by the ammonia mixture in being vaporized. The amount is, of course, approximately equivalent to the amount given up by the substance being cooled in the refrigerator. On the scale of abscissas, it will be found that the total heat of the ammonia mixture at the point *B* is 139 B.t.u. per pound and that the total heat at the point *D*, which represents total vaporization of ammonia, is 613 B.t.u. per pound. The heat transfer in the cooling coils of the evaporator during the vaporization of the ammonia mixture from the quality of 0.16 to complete vaporization is 613 − 139 or 474 B.t.u. per pound of ammonia.

In the next step of the compression refrigerating cycle, the ammonia vapor is taken from the cooling coils of the evaporator through the suction pipe into the cylinder of the compressor where the ammonia vapor is compressed approximately adiabatically, without much gain or loss of heat to the higher pressure of the system or 169 pounds per square inch absolute. The line on the chart representing this adiabatic compression must, of course, be parallel to the constant-entropy lines shown in the right-hand portion of the chart and is represented by the line *DE*. The point *E* is naturally found at the intersection of the constant-entropy line through *D* with a horizontal constant-pressure line through the point *A*. The point *E* represents the condition of the ammonia vapor after being compressed adiabatically in the compressor. The heat equivalent of the work done in the compressor per pound of ammonia vapor handled is, of course, the difference between the total heats measured on the scale of abscissas at the points *D* and *E*. These values are, respectively, 613 and 713, the difference being 100 B.t.u. per pound of ammonia (see pp. 104 and 273).

The condition of the ammonia vapor, as represented by any points in that portion of the chart to the right of the saturated-vapor line, is superheated, meaning the temperature is higher than the saturation temperature corresponding to the pressure. In Fig. 200, the point *D* is on the saturated-vapor line indicating that its condition is dry and saturated. The point *E* is in the region to the right of the saturated-vapor line and is, therefore, *superheated*. The amount of superheat is indicated by the lines

of constant temperature, which show that at the point E the superheat is a little more than 210° F.

In the compression system, after the superheated vapor is discharged from the compressor it passes into the condenser, where heat is removed by cold water used for cooling. In this cooling process, all the ammonia vapor is condensed; in other words, the ammonia vapor changes from the superheated condition at the point E in the chart to the condition at A at the same pressure (169 pounds per square inch absolute), where it is all liquid. The heat removed in the condenser from the ammonia vapor is 713 − 139 or 574 B.t.u. per pound of ammonia.

The lines AB, BD, DE, and EA represent, as laid out in Fig. 200, the *dry-compression system* of refrigeration for the so-called *standard conditions* of 86° F. for the upper and 5° F. for the lower limit of temperature of the ammonia.

The region in the diagram to the right of the saturated-vapor line represents *superheated* ammonia vapor. Small amounts of superheat are near the saturated-vapor line, and, as the amount of superheat in the ammonia vapor increases, the space from this line increases toward the right. The chart shows, also, constant-volume, constant-entropy, and constant-temperature lines.

Pressure-total Heat Chart for Superheating and Aftercooling.—The pressure-total heat chart, as used in Fig. 200, shows a cycle of refrigeration for dry compression with no superheating or aftercooling. The same chart can be used to show, also, refrigerating cycles in which there is superheating of the ammonia vapor between the cooling coils of the evaporator and the suction valve of the compressor in which the liquid ammonia from the condenser is aftercooled before expansion takes place in the expansion valve.

Superheating of the ammonia vapor which comes from the cooling coils of the evaporator before it reaches the suction side of the compressor makes some modification of the simple refrigerating diagram as shown in Fig. 200. The dotted lines at the right-hand side of the figure show the modification resulting from superheating the ammonia vapor before it enters the suction side of the compressor. These dotted lines join the points A, B, F, and G. In this "vapor mixture" region, the line AB represents, as before, a constant heat line of the ammonia through the expansion valve; the line BD represents the evaporation of the ammonia liquid in the cooling coils of the evaporator as it absorbs

heat from the substance cooled in the refrigerator; the line *DF* represents the heat added to the dry and saturated ammonia vapor so that, because of superheating, its temperature is above 5° F.; the line *FG* indicates the adiabatic compression of the superheated ammonia vapor in the compressor; the line *GA* shows the loss of heat by the ammonia vapor cooling in the condenser, reducing the temperature of the superheated ammonia vapor from 240° F. to a completely liquefied state at the point *A*.

Aftercooling is illustrated in the same figure by the addition of the dotted lines *AH*, *HK*, and *KB* to the refrigerating cycle. The line *AH* shows the cooling (aftercooling) of the *liquid ammonia* at the pressure corresponding to the standard temperature of 86° F. at which it leaves the condenser to the temperature of 60° F. (represented by point *H*). Constant heat expansion through the expansion valve is shown by the vertical line drawn through *H* to intersect the extension of the line *BD* at the point *K*.

The refrigerating cycle for dry compression with no superheating of the ammonia vapor as it leaves the compressor, but with aftercooling by which the temperature of the liquid ammonia is reduced from 86 to 60° F., is shown by the lines joining the points *H*, *K*, *D*, *E*, and *H*.

Wet-compression Cycle.—When enough liquid ammonia is taken into the compressor through the suction pipe to absorb all the heat generated by the compression of the ammonia vapor, the refrigerating cycle is called *wet compression*. *Theoretically*, enough liquid ammonia should be mixed with the ammonia vapor which enters the compressor so that the latter will be dry and saturated at the end of compression. This means that if, for example, the refrigerating cycle begins at the point *A* (Fig. 200), with liquid ammonia (without aftercooling), and has constant heat expansion along the vertical line from *A* to *B* and further evaporation at constant pressure due to the absorption of heat along the horizontal line through *B*, the cycle of refrigeration must pass through the point *M* located on the horizontal line through *A*, in order that the ammonia vapor may be dry and saturated at the end of the adiabatic compression. Now, if the end of the adiabatic compression is at *M*, the beginning of the compression must be at some point on the horizontal line through *B*. *Adiabatic compression* means, of course, a compression that is represented on a line of constant entropy. Such a line is shown by *MN*.

A cycle of wet compression may be shown by the lines joining the points A, B, M, and N, where the evaporation in the cooling coils of the evaporator is indicated by BN; adiabatic compression in the compressor by NM; and condensation of the ammonia

TABLE IX*b*.—FUNDAMENTAL CONSTANTS OF AMMONIA

	Dry compression with aftercooling and superheating	Dry compression with liquid cooled to 60° F.	Dry compression with vapor superheated to 40° F.	Dry compression with liquid aftercooled to 60° and vapor heated to 40° F.	Wet compression with no aftercooling or superheating
Temperature in evaporator	5°	5°	5°	5°	5°
Pressure in evaporator	34.3	34.3	34.3	34.3	34.3
Specific volume of vapor, to compressor	8.2	8.2		8.9	7.24
Temperature in condenser	86°	86°	86°	86°	86°
Pressure in condenser	169	169	169	169	169
Temperature after compression	210°	210°	259°	259°	86°
Specific volume of gas after compression, cubic feet	2.36	2.36	2.58	2.58	1.77
Heat content of superheated vapor, B.t.u. per pound	712.9	712.9	742.5	742.5	631.5
Heat content of vapor from evaporator, B.t.u.	613.4	613.4	634.0	634.0	550.5
Heat equivalent of work of compression, B.t.u.	99.5	99.5	108.5	108.5	81.0
Heat content of liquid in condenser, B.t.u.	138.9	109.3	138.9	109.3	138.9
Heat rejected in condenser (item 8– item 11), B.t.u.	574	603.6	603.6	633.2	492.6
Refrigerating effect (item 9– item 11), B.t.u.	474.5	504.1	495.1	524.7	411.6
Quality of mixture after expansion	0.160	0.108	0.160	0.108	0.160
Pounds of ammonia per minute per ton of refrigeration	0.4216	0.3970	0.4040	0.3811	0.4862
Theoretical volume of ammonia per ton per minute, cubic feet	3.457	3.257	3.596	3.392	3.52
Theoretical horsepower per ton	0.988	0.931	1.033	0.974	0.928
Coefficient of performance	4.768	5.070	4.563	4.841	5.080
Quality of mixture before compression					0.890

vapor in the condenser by MA; so that the ammonia vapor will be dry and saturated at the end of the compression. For practical reasons, however, it is not desirable in a wet compression system to have so much liquid ammonia in the mixture as would

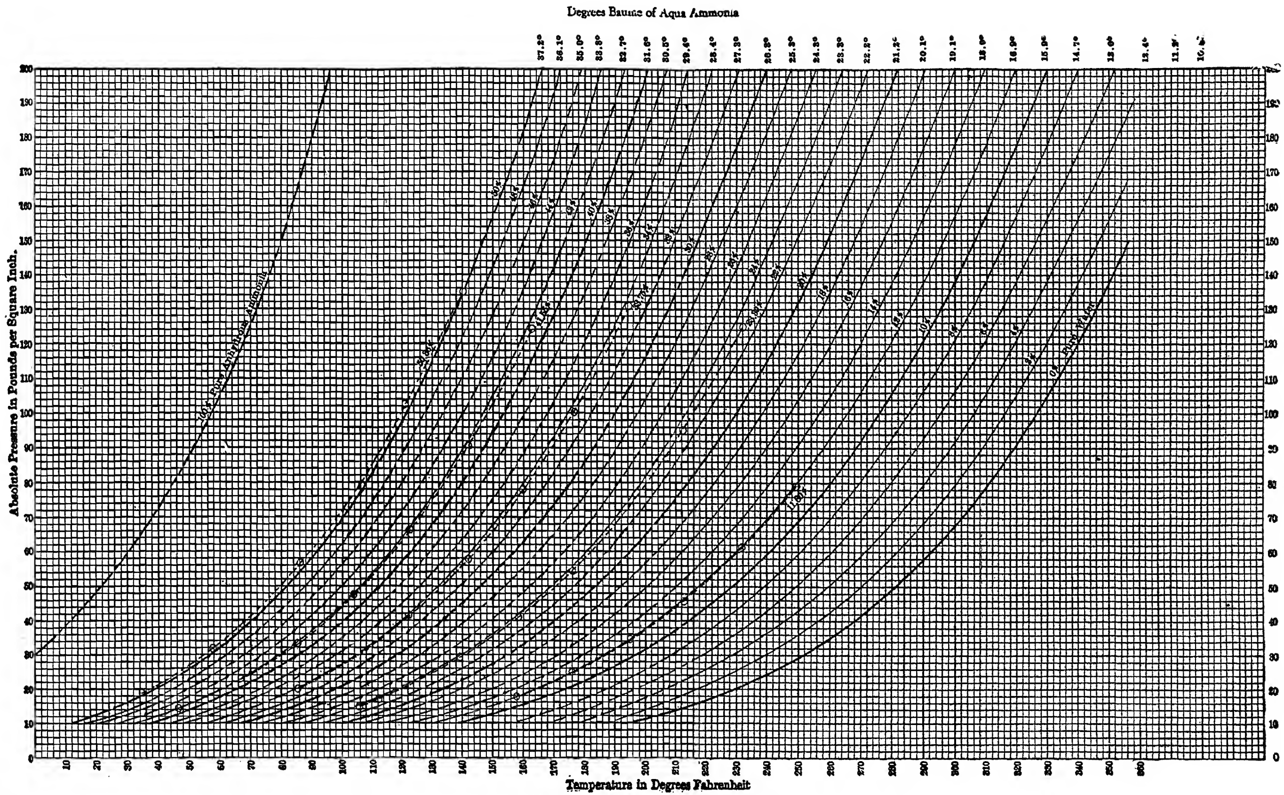


FIG. 201.—Pressure, temperature and concentration of aqua ammonia.

be required for this theoretical condition, so that, as a general rule in practical work, the ammonia vapor at the end of compression will be slightly superheated instead of being dry and saturated, a practice considered much better. All the refrigerating cycles which have been illustrated on the pressure-total heat diagram have been based on the *standard* conditions of operating pressures corresponding to the saturation temperatures of 86° F. in the condenser and 5° F. in the cooling coils of the evaporator. Methods of representation on such charts would be similar, however, for other temperature or pressure conditions.

Table IX*b* compares the effect of dry and wet compression for the standard condition of 5° F. evaporator temperature and 86° F. condenser temperature. It will be well to note the results of after-cooling and superheating the ammonia vapor.

Ammonia Absorption System of Refrigeration.—The absorption system of refrigeration depends on the fact that anhydrous ammonia has the property of forming aqua ammonia. The amount of ammonia which can be absorbed by water depends on the temperature of the water; the colder the water the greater its absorption of ammonia.

Heat Properties of Ammonia Solutions.—From the above discussion, it is apparent that an understanding of the thermodynamic properties of ammonia solutions is necessary in order thoroughly to understand the principles of the absorption system.

It has long been known that anhydrous ammonia has a great affinity for water, and a solution thus formed is said to have a "concentration of 30 per cent" if the solution contains 30 per cent ammonia and 70 per cent water by weight.

The temperature at which an ammonia solution will boil when under pressure and of a definite concentration has been studied by Mollier. Macintire has given the following equation from which the boiling temperature can be determined:

$$\frac{T_1}{T_2} = 0.00471z + 0.655$$

where T_1 = temperature of saturated ammonia corresponding to the pressure, degrees Fahrenheit, absolute

T_2 = boiling temperature, degrees Fahrenheit, absolute

z = per cent concentration

A family of curves, as shown in Fig. 201, has been arranged to simplify the use of the above equation.[1]

[1] See Marks, "Handbook of Mechanical Engineering," Fig. 2, p. 1823.

It has been previously stated that in the generator there is a mixture of water and ammonia vapor. The total pressure in the generator is the sum of the partial pressures of ammonia and water vapor. The partial pressure of the water vapor has been taken by Professor Spangler to be the steam pressure at the temperature considered multiplied by the ratio of the number of molecules of water in a certain amount of solution to the total number of molecules of the solution. Hence, it follows that

$$\frac{z}{17} = \text{relative number of ammonia molecules}$$

$$\frac{100 - z}{18} = \text{relative number of water molecules}$$

for which the partial steam pressure p_a, in pounds per square inch absolute, is

$$p_a = \frac{\frac{100 - z}{18}}{\frac{100 - z}{18}} = p\frac{1{,}700 - 17z}{1{,}700 + z}$$

where p equals the absolute steam pressure at the given temperature in pounds per square inch.

When 1 pound of ammonia vapor is absorbed by 200 pounds of water, about 893 B.t.u. of heat are developed. If a greater weight of water is used, the number of absorbed heat units is the same. For this reason, this value (in B.t.u.) is called the *heat of complete absorption.* On the other hand, if a smaller weight of water is used, less heat is developed. This latter case is called the *heat of partial absorption,* because, if more water is added to produce a dilution of 1 in 200, the remaining heat (to make the total 893 B.t.u.) would be generated.

It was found by Berthelot that the heat of complete dilution (H_d) expressed is 142.5 times the weight of the ammonia (in pounds) in the solution per pound of water. This value is expressed by the equation

$$H_d = 142.5 \frac{}{100 - z}$$

in B.t.u. per pound of ammonia solution having a concentration of z per cent, where z is the percentage of concentration of the solution which is formed.

The partial heat of absorption (H_a) is, then,

$$H_a = 893 - 142.5 \; \frac{}{100 - z}$$

in B.t.u. per pound of ammonia solution having a concentration of z per cent.

If w pounds of ammonia are absorbed, the partial heat of absorption for this amount is

$$H_a' = w\Big[893 - 142.5 \cdot$$

in B.t.u. per pound of ammonia solution having a concentration of z per cent.

In the absorber, heat is generated by the addition of ammonia to the ammonia solution. If the strength of an ammonia solution is changed from z to z', this heat which is generated is given by the following equation:

$$_{z-z'} = 893 - \qquad - z' \qquad 100$$

in B.t.u. per pound of ammonia vapor which is added.

According to the experiments of Mollier, the heat of solution developed in changing a weak solution having a concentration of z per cent to one having a concentration of z' per cent depends upon the mean concentration $x = \dfrac{z + z'}{2 \times 100}$. The heat generated then is given by the equation $H_{z-z'} = 345\ (1 - x) - 400x^2$ in B.t.u. per pound of liquid ammonia which is added.

In analyzing the operation of the absorption system, it is customary to determine the amount of strong-ammonia solution in circulation per pound of anhydrous ammonia. If the following symbols are used, suitable equations for this determination can be found: W_w = weight of weak ammonia solution per pound of anhydrous ammonia, pounds; W_s = weight of strong ammonia solution per pound of anhydrous ammonia, pounds; Z_s = percentage of concentration of strong ammonia solution; Z_w = percentage of concentration of weak ammonia solution.

By equating the total weight of aqua ammonia entering the generator to the total weight leaving it,

$$W_s = W_w + 1$$

Again, by equating the weight of anhydrous ammonia entering the generator to the weight of anhydrous ammonia leaving it,

$$Z_s \times W_s = 1 + Z_w \times W_w$$

Solving the two equations above for the weight of weak ammonia solution,

$$W_w = \frac{1 - Z_s}{Z_s - Z_w}$$

The calculations may be simplified by reference to the table on page 295. From a thermodynamic viewpoint, it is interesting to calculate the heat balance of the absorption system, but since little is known as to all of the thermodynamic properties of ammonia solutions, it is difficult to make these calculations with much accuracy.

The following equation expresses the *heat balance* for the absorption system:

where

H_g = heat imparted to the fluid in the generator per pound of anhydrous ammonia passing through the expansion valve, B.t.u. per pound.

H_2 = heat absorbed in the cooling coils of the evaporator, B.t.u. per pound of anhydrous ammonia.

H_3 = heat rejected to the cooling water of the condenser, B.t.u. per pound of anhydrous ammonia.

H_4 = heat withdrawn from the absorber, B.t.u. per pound of anhydrous ammonia.

H_5 = heat equivalent of work of the pump, B.t.u. per pound of anhydrous ammonia.

H_6 = heat loss of radiation, etc., B.t.u. per pound of ammonia.

The heat H_2 absorbed in the coils of the evaporator can be calculated from the equation $H_2 = H_1'' - h_2$, where H_1'' is the heat content in B.t.u. per pound of the ammonia vapor corresponding to the pressure in the cooling coils or evaporator and h_2 is the heat content of the liquid ammonia in B.t.u. per pound corresponding to the temperature of the liquid ammonia entering the expansion valve.

The heat rejected to the cooling water by the condenser can be obtained if the temperature and pressure of ammonia vapor

entering and the temperature of the liquid ammonia leaving the condenser are known. Then, $H_3 = H_2'' - h_2$, where H_2'' is the heat content in B.t.u. per pound in the ammonia vapor entering the condenser and h_2 is the heat of the liquid in B.t.u. per pound of the liquid ammonia leaving the condenser.

TABLE X.—POUNDS OF STRONG AQUA AMMONIA REQUIRED PER POUND OF AMMONIA EVAPORATED

Concentration of weak aqua ammonia, per cent	Concentration of strong aqua ammonia, per cent								
	20	22	24	26	28	30	32	34	36
18	41	20.5	13.67	10.25	8.2	6.84	5.86	5.12	4.55
20	..	40	20	13.33	10	8.0	6.67	5.71	5.0
22	..	....	39	19.5	13	9.75	7.8	6.5	5.57
24	..	....	.....	38	19	12.67	9.5	7.67	6.32
26	..	....	.....	.....	37	18.5	12.33	9.25	7.4
28	..	....	.....	.....	.....	36	18	12.0	9.0
30	..	....	.....	.....	.....	.....	35	17.5	11.67

The heat withdrawn from the absorber H_4 can be found from the following: Heat must be removed from the weak aqua ammonia which enters the absorber at the temperature t' and is cooled to the temperature of the strong aqua ammonia leaving the absorber t_0; heat is removed due to the heat of solution $H_{\text{sol.}}$, to which must be added an amount of heat equal to the total heat of the incoming ammonia vapor minus the heat of the liquid at the temperature of the strong aqua ammonia leaving the absorber.

The heat of solution has already been considered (p. 293) and may be found from the equation

$$H_{\text{sol.}} = 345(1 - x) - 400x^2$$

Hence,

$$= H_{\text{sol.}} + W_w(t' - t_0) + H_a -$$

in which H_a is the total heat at the absorber pressure and $h_{t'}$ is the heat of the liquid at the temperature of the leaving strong aqua ammonia.

The pump transfers W_s pounds of strong aqua ammonia at the pressure P_1 in the absorber to the pressure P_2 in the generator. If V is the volume of 1 pound of strong aqua ammonia, then

$$\overline{778}$$

The heat loss by radiation H_6 will be about 5 or 10 per cent of the heat supplied to the generator H_1.

Heat of Liquid for Mixtures of Ammonia and Water.—The heat added to 1 pound of an ammonia solution can be determined by assuming that the heat capacity of a solution of ammonia

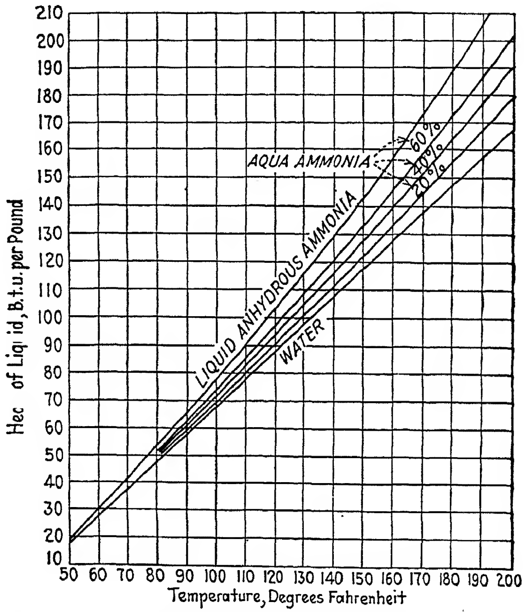


FIG. 202.—Heat of liquid for water and anhydrous ammonia and aqua ammonia of various concentrations.

and water is the sum of the heat of liquid of the constituents according to their proportions in the solution. If h_a and h_w are the heat of liquid (as found in the ammonia and steam tables) and z is the per cent of concentration, the weight of water is $\frac{100 - z}{100}$ per pound of solution, and the heat of the liquid of an ammonia solution is then equal to

$$\frac{}{100} \qquad \frac{(100 - z)}{100}$$

Values of the heat of the liquid for ammonia solutions for various temperatures and concentrations can be found graphically[1] from Fig. 202. The heat values given in Fig. 201 are determined from 32° F.

Heat Supplied to Generator.—The heat supplied to the generator may be found from the heat balance by assuming the radiation loss, or an approximation can be made by considering that the generator is an absorber operating in a reverse process. The heat transferred in the generator per pound of anhydrous ammonia is the sum of heat added to the strong ammonia solutions between the temperature of the entering strong ammonia solution and the temperature of the weak ammonia solution leaving the generator, the heat of solution, and the ammonia vapor from the liquid to the superheated condition. Expressing this statement in the form of an equation,[2] we have the following:

$$H_g = W_w(h_2' - h_1') + h_{\text{sol.}} + (H_1 - h_0)$$

where H_g = Heat supplied by generator, B.t.u. per pound

W_w = Weight of weak ammonia solution per pound of anhydrous ammonia pounds

h_2' = Heat of liquid of ammonia solution of z concentration for the temperature of the weak solution leaving generator, B.t.u. per pound

h_1' = Heat of liquid of ammonia solution of z' concentration for the temperature of the strong solution entering generator, B.t.u. per pound

H_1 = Total heat of ammonia vapor in generator, B.t.u. per pound

h_0 = Heat of liquid ammonia corresponding to temperature of strong ammonia solution, B.t.u. per pound

Values for the heat of solution $h_{\text{sol.}}$ of liquid ammonia may be taken from the table shown on page 298.

Density and Specific Gravity of Ammonia Solutions.—In order to determine the horsepower developed by the strong aqua-ammonia pump, it is necessary to know the density of the solution. This is determined from its specific gravity, and the relationship is shown graphically[3] in Fig. 203. The specific

[1] See POWER, "Practical Refrigeration," Fig. 101.

[2] The equation does not consider the effect of the water vapor in the generator.

[3] See POWER, "Practical Refrigeration," Fig. 99.

gravity is fixed at a temperature of 60° F. At other temperatures, the solution will occupy a different volume, and, therefore, the specific gravity will change even when the concentration remains the same.

If the temperature is within 5 to 10° F. of 60° F., the percentage of concentration can be taken from the curve when the specific gravity has been found. For the high temperatures found in an

TABLE XI.—HEAT OF SOLUTION OF LIQUID AMMONIA

B.t.u. given up per pound of ammonia absorbed

Concentration[1]	Heat of solution, B.t.u. per pound	Concentration[1]	Heat of solution, B.t.u. per pound	Concentration[1]	Heat of solution, B.t.u. per pound	Concentration[1]	Heat of solution, B.t.u. per pound	Concentration[1]	Heat of solution, B.t.u. per pound	Concentration[1]	Heat of solution, B.t.u. per pound
0	347.4	11	302.8	21	253.8	31	197.6	41	135.0	51	63.0
1	343.8	12	298.2	22	248.4	32	191.9	42	127.8	52	55.8
2	340.2	13	293.6	23	243.0	33	186.1	43	120.6	53	48.6
3	336.6	14	289.0	24	237.6	34	180.4	44	113.4	54	41.4
4	333.0	15	284.4	25	232.2	35	174.6	45	106.2	55	34.2
5	329.4	16	279.4	26	226.4	36	168.1	46	99.0	56	27.4
6	325.0	17	274.3	27	220.7	37	161.6	47	91.8	57	20.5
7	320.6	18	269.2	28	214.9	38	155.2	48	84.6	58	13.7
8	316.2	19	264.2	29	209.2	39	148.7	49	77.4	59	6.8
9	311.8	20	259.2	30	203.4	40	142.2	50	70.2	60	0.0
10	307.4										

[1] Average concentration, per cent of ammonia by weight.

absorption system, there is usually difficulty in obtaining the concentration by testing. This is due to the volatility of the sample of the ammonia solution because of its high temperature and low pressure when it is removed from the system. In order to take a sample for testing, it is customary to have suitable outlets arranged at points where the temperatures are comparatively low, as on the discharge side of the pump for the strong ammonia solution and between the absorber and weak-liquor cooler for the weak ammonia solution. A glass graduate and hydrometer are needed for testing. The graduate should be half filled with water cooled to about 32° F. The tube for obtaining the samples of the ammonia solutions should be placed so that its outlet is below the surface of the water in the graduate. The ammonia solution to be tested should be discharged into the graduate until the temperature of the mixture rises to 60° F. The specific

gravity then can be found by means of the hydrometer. Sinc it is the concentration of the sample that is desired, one can then calculate the concentration by equating the weight of ammonia

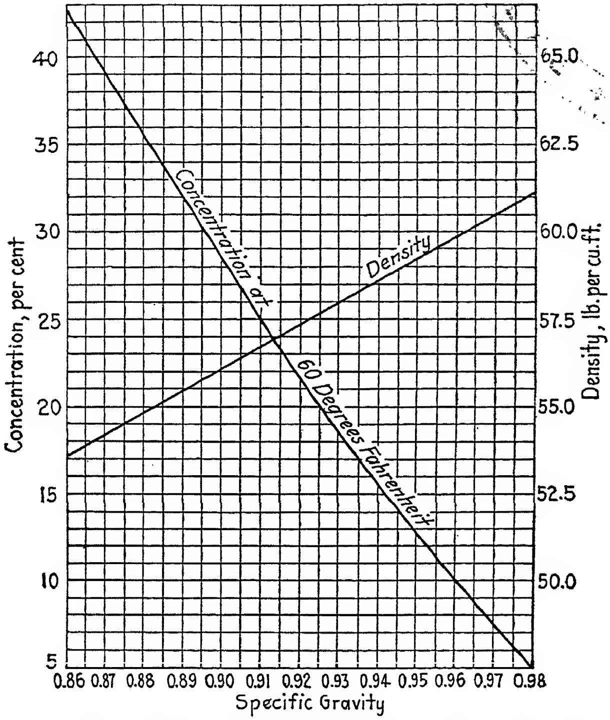


FIG. 203.—Density and specific gravity of aqua ammonia.

in the solution to the total weight of ammonia in the final mixture of ammonia and water, as follows:

$$S_p \times Z(1 - R) = 1 \times S_{pm} \times Z_m$$

or

$$(1)$$

Equating the weight of the mixture of the sum of the weights of the ammonia solution and water, the following relation is obtained:

$$1 \times S_{pm} = (R \times 1) + (1 - R)S_p \qquad (2)$$

Eliminating S_p (which is not known) between equations (1) and (2) the following expression is obtained for the desired concentration Z.

$$Z = Z_m \div \left(1 - \frac{R}{\alpha}\right.$$

where Z = concentration of ammonia solution, per cent
Z_m = concentration of ammonia and water mixture, per cent
R = ratio of the volume of cold water before mixing with sample to the volume of the ammonia and water mixture.
S_p = specific gravity of the ammonia solution tested.
S_{pm} = specific gravity of the ammonia and water mixture.

If the temperature is higher than 60° F., the specific gravity will be somewhat lower, between about 0.001 and 0.005 for each 10° F. of temperature difference, depending on the concentration. The strong solutions have a higher coefficient of expansion and, therefore, need a greater correction factor than the weak solutions. The specific gravity at 60° F. can then be found from the following formula:

$$S_{p60} = S_{pt} + 0.003(t - 60)(1 -$$

where S_{p60} = specific gravity at 60° F.
S_{pt} = specific gravity at the temperature t

The *approximate* specific gravity of an ammonia solution S_{pa}, taking the specific gravity of water as unity, can be found from the following formula:

$$\frac{}{1{,}000}\left(Z - \frac{}{100} + \right.$$

where Z is the percentage of concentration of the ammonia solution.

CHAPTER VIII

REFRIGERATION ECONOMICS AND PLANT TESTING[1]

Economics in Refrigeration.—This chapter is devoted to the application of the general principles that give the most economical operation of refrigerating plants. The study must include a comparison of actual operating results with ideal performance. Obviously, it is the duty of those in charge of refrigerating plants to produce the maximum amount of refrigeration with a minimum expenditure of labor, materials, and mechanical or electrical energy. In the final analysis, the economics of any engineering process is concerned with the conditions of value, price, cost, and profit.[2]

Refrigeration Costs.—A number of factors must be considered in estimates of the cost of refrigeration, because the estimates will depend on such items as the geographical location; the cost of fuel, labor, and supplies; the size of the plant; and the efficiency of the mechanical equipment. These general items may be divided into classifications under the general headings of

a. Fixed Charges.—These are independent of the refrigeration output of the plant and go on whether or not the plant is operated.

b. Operating Expenses.—These expenses depend on the amount of output; in other words, they increase, in some measure, in proportion to the increase in the tons of refrigeration of the plant.

Somewhat in detail, the classifications may be tabulated as shown at top of p. 302.

[1] The test code for refrigerating systems adopted by the American Society of Refrigerating Engineers as well as the codes of the American Society of Mechanical Engineers for testing steam engines, steam turbines, and internal-combustion engines are given in "Power Plant Testing," by James A. Moyer, McGraw-Hill Book Company, Inc., New York.

[2] *Value* is the exchangeable worth of property or service. *Price* is the money given for property or service. *Cost* is the actual money expended for property or service. It includes not only the price paid at the time a property is acquired but also the other items of expenditure which are chargeable in determining the present value of the property. Profit is the difference between the selling price and the cost.

Fixed Charges	Operating Expenses
Interest	Power
Depreciation	Labor
Repairs	Ammonia or other refrigerant
Taxes	Oil
Insurance	General supplies
Incidentals	

Fixed charges and operating expenses will vary considerably with the operating conditions. By keeping the mechanical equipment of the plant in good condition, its depreciation may be reduced a great deal, and, at the same time, there will be a saving in the operating expenses for such items as labor, maintenance, etc. A refrigerating plant which is operated on a commercial basis should, of course, have sufficient profits for the depreciation charges, so that when the equipment is no longer serviceable, it may be replaced with new machinery.

Fixed charges do not vary a great deal in different parts of the country, except, of course, that the total of interest charges is very much larger where expensive land must be purchased than where relatively cheap land is available. Interest and other fixed charges will be about as follows:

Fixed Charges	Per Cent
Interest	6 to 7
Depreciation	5 to 6
Repairs	3 to 5
Taxes and insurance	2 to 4
Total	16 to 22

It will be noted that the total amount of interest and other fixed charges varies from 16 to 22 per cent of the original investment, which includes the total cost of the plant and all the items of labor, material, interest, engineering, and overhead charges during construction.

Cost of Power.—Until recent years, nearly all refrigerating plants were operated by steam engines, simple non-condensing, slow-speed Corliss engines being used for plants having capacities of 20 to 200 tons of refrigeration per day. Larger plants having capacities up to 1,000 tons of refrigeration per day are operated with compound condensing steam engines which receive their steam from boilers equipped with automatic stokers. In some modern plants, electric motors are used to drive the mechanical

equipment, the electricity being purchased from a company distributing and selling electric power.

Small refrigerating plants operated by steam will produce only about 10 tons of refrigeration per ton of coal, while some of the largest plants will produce about twice as much per ton. The cost of coal varies, of course, a great deal with the location of the plant, particularly as the distance varies for the transportation of the coal from the mines. The cost of electric current for refrigerating plants is not nearly so variable as that of steam power. The average rate charged by large electric companies for electric current in ice-making and refrigerating plants varies from 0.7 cent to 1.4 cents per kilowatt-hour. The average is, therefore, approximately 1 cent per kilowatt-hour.

Instruments for Refrigeration-plant Testing.—Thermometers and pressure gages must be installed at points where it is desirable to determine the conditions of pressures and temperatures. These instruments are very helpful in maintaining the system in proper operation. Gage glasses should be provided at all points of the system where it is desirable to ascertain the level of the working liquids. Suitable connections for testing aqua ammonia or anhydrous liquid into the system should be provided. Likewise, suitable connections should be provided for withdrawing strong and weak liquor for testing purposes.

Use of the Indicator.—In making a test on a compressor, it is necessary to determine the amount of power required to compress the ammonia vapor. At the same time, it is often necessary to find the amount of power developed in the cylinder of the steam or gas engine which drives the compressor. These measurements are most conveniently obtained with an instrument called an *indicator*, which gives also a means of finding out the action of the compressor valves and the manner in which the vapor of the refrigerant is being compressed. The indicator serves as a means to draw on a card of paper attached to the instrument a diagram representing the pressure in the cylinder for every position of the piston for both the compression and suction strokes. Such an indicator diagram shows a complete record of what takes place in the cylinder for one or more revolutions.

Figure 204 shows the construction of an outside spring indicator. It consists of a cylinder C containing a nicely fitted piston B, which is free to move up and down without appreciable friction. To the piston is connected a piston rod R having a

double coil spring S connected to its upper end. The piston rod is also connected to the pencil mechanism, which consists of levers and a pencil arm L the right-hand end of which carries a pencil or a brass stylus, moving vertically parallel to the axis of the indicator drum D. The drum is given a to-and-fro rotary motion, coincident with, and bearing a constant ratio to, the motion of the piston of the engine or compressor. It is moved

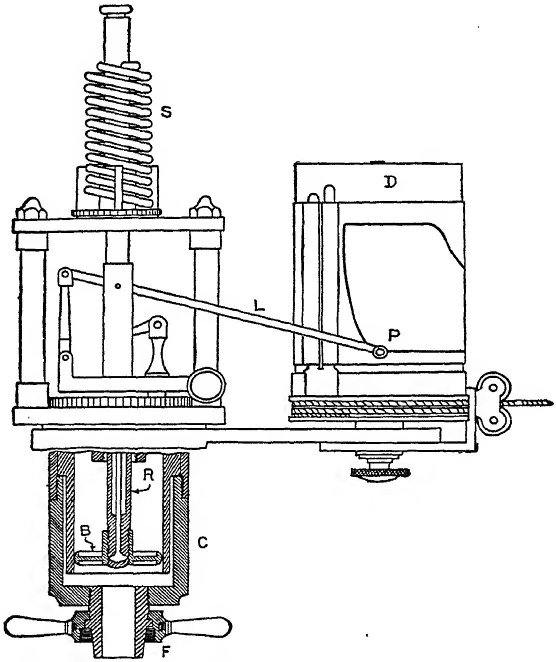


FIG. 204.—Typical indicator for testing ammonia compressors.

in one direction by means of a cord attached indirectly to the crosshead of the engine and wound around the base of the drum. Inside the drum is a coil spring, which is wound up when the cord is pulled out and, in unwinding, when the pull on the cord is removed, turns the drum in the opposite direction. The cylinder C is connected to the clearance space of the compressor cylinder by means of a union F and a short piece of pipe containing an indicator cock. The varying pressure of the ammonia vapor in the cylinder of the compressor acts against the underside of the

piston of the indicator and pushes it up against the action of the spring S.

The springs used with indicators are made in different sizes. In order to obtain a correct diagram, the movement of the pencil point must exactly represent a certain pressure. Springs are marked according to the movement of the pencil point; *i.e.*, a No. 60 spring will produce a pencil movement of 1 inch when a pressure of 60 pounds per square inch acts against the piston. A pressure of 90 pounds per square inch will cause a pencil movement (vertical) of 1½ inches.

The back-and-forth motion of the indicator drum D is in unison with the motion of the piston of the compressor. Also, the pencil point P moves up and down with the varying pressure in the cylinder of the compressor. The combining of these two motions gives the indicator diagram traced by the pencil P, which shows what is taking place inside the cylinder of the compressor. The diagram traced by the pencil is an area representing the work being developed in the cylinder of the compressor. This area represents work, because the pressure exerted on the piston varies with that in the compressor cylinder, and the back-and-forth motion of the drum is proportional to the piston travel. When using an indicator attached to the cylinder of a steam engine, the indicated horsepower developed by the engine can be calculated. This is the power which is being developed in the cylinder of the steam engine.

Indicator Diagram.—An indicator diagram obtained from an ammonia compressor is shown in Fig. 205. The length of the diagram A represents the stroke of the compressor according to some scale. At the end of the suction stroke or the point E, the cylinder is filled with ammonia vapor; when the piston of the compressor starts on its compression stroke, the ammonia vapor in the cylinder is compressed along the line EG. At the point G, the ammonia vapor has been compressed above the pressure which opens the discharge valves and is then forced out of the cylinder during the remainder of the stroke GJ.

As already mentioned, the ammonia vapor is compressed somewhat above the discharge pressure before the discharge valve opens, due to the fact that the discharge valve is conical in shape and that the side next to the discharge chamber is somewhat larger than the side next to the cylinder. The discharge valve is forced to its seat by the pressure of the vapor in the condenser and opens against this pressure.

In Fig. 205, the ammonia vapor is compressed along the line EF, which is for dry compression. If the compressor is operating with wet compression, the ammonia vapor would be compressed along the line EH. Since the area of an indicator diagram represents work expended in compressing the ammonia vapor, it can be seen that more work is required to compress the ammonia vapor for dry than for wet compression.

After the compressed vapor is forced out of the cylinder, the discharge valve closes, and the piston starts back on its suction stroke. At the beginning of the suction stroke, there is left in

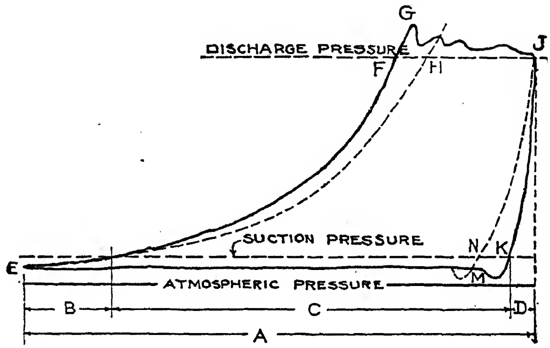


FIG. 205.—Typical indicator diagram of ammonia compressor.

the clearance space some ammonia vapor which expands along the line JM until the pressure falls slightly below the suction pressure. The suction valve opens at the point M, and new vapor is drawn into the cylinder during the remaining portion of the suction stroke ME. The dip in the reexpansion line JK is due to the fact that the inner face of the suction valve is larger than the outer face and, also, because of the tension of the spring on the valve stem.

As seen from Fig. 205, the piston must move a distance B on the compression stroke before the pressure rises above the suction pressure. Also, the piston must move a distance D on the suction stroke before the ammonia vapor left in the clearance space expands to the suction pressure. It can now be seen that, instead of a cylinder full of new vapor being drawn in during each suction stroke, there is only a volume equal to C, measured at the suction pressure. The actual capacity of the compressor is, then, only C, while the theoretical capacity is the full piston displacement

A. The loss of capacity B is due to the action of the suction valve, while that of D is due to the clearance space. If the clearance space had been larger, the reexpansion line would have been along some line, as JN. The effect of a larger clearance space would be to reduce the actual capacity of the compressor still more. The ratio of the length C on the diagram to the length A, or $C \div A$, is called the apparent *volumetric efficiency* of the compressor. This is the ratio which the actual useful volume of vapor taken into the cylinder bears to the full piston displacement. The heavy, full lines are drawn by the indicator and also the atmospheric line. The latter, representing the atmospheric pressure, is drawn by placing the pencil to the indicator card after the indicator cock is closed. It is useful as a reference line by which other pressures may be measured.

Mean Effective Pressure.—To find the horsepower required to compress the ammonia, it will be necessary to obtain the average pressure during a complete revolution of the compressor. This is obtained from the indicator diagram. The average pressure is equal to the average height of the diagram between the suction line and the compression and discharge lines times the scale of the spring. The average pressure is called the *mean effective pressure* (*or m.e.p.*). The mean effective pressure can be found from the indicator diagram by dividing it into 10 equal parts and measuring the height of the diagram at these divisions. The sum of these different heights divided by 10 gives the mean height. This value multiplied by the scale of the spring gives the mean effective pressure. A very easy way to divide the diagram into 10 equal parts is shown in Fig. 206. To do this, draw two lines perpendicular to the atmospheric line and through the extreme points of the diagram. Now lay an ordinary rule in such a position that the zero mark is on the left-hand perpendicular line. Place the 5-inch mark on the perpendicular at the right-hand end of the diagram. From the first ¼-inch mark, draw through the diagram a perpendicular line to the atmospheric line, and draw lines from each ¾- and ¼-inch mark to the 5-inch mark. The reason for dividing the diagram at the ¼-inch mark is to prevent the first and last divisions from coming at the ends of the diagram, for here the pressure is varying rapidly.

Another way of obtaining the mean effective pressure is by the use of a planimeter, an instrument which measures an area directly in square inches. Obtain the area of the indicator dia-

gram and divide this area by the length of the diagram in inches, giving the mean height in inches, which, multiplied by the scale of the spring, gives the mean effective pressure.

Indicated Horsepower.—The indicated horsepower (or i.h.p.) can be found from the following formula:

$$\text{Indicated horsepower} = \frac{p \times l \times a \times n}{33{,}000}$$

where p = mean effective pressure, pounds per square inch
l = length of the stroke, feet
a = area of the piston, square inches
n = number of revolutions per minute

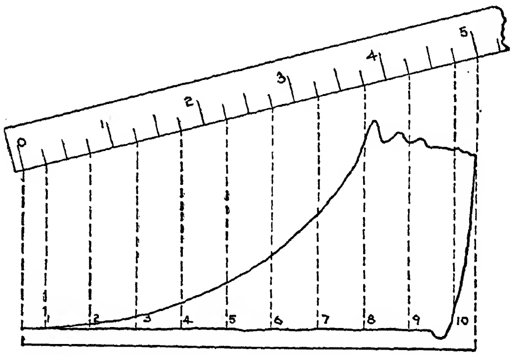


Fig. 206.—Method of calculating mean effective pressure of indicator diagram.

When using the above formula, the m. e. p. is measured from the indicator diagram as obtained from one end of the cylinder. Where the compressor is double-acting, a diagram should be taken for each end. The indicated horsepower should be calculated for each end independently, and the total indicated horsepower of the compressor is obtained by adding these values.

With double-acting compressors, the entire piston area of the crank end is not subjected to the full pressure, because the piston rod reduces the effective area.

To illustrate the method of calculating the indicated horsepower of a compressor, suppose the combined heights of the 10 divisions of the indicator diagram to be 7.0 inches, and the diagram to have been drawn with a No. 80 spring. The average height of the diagram is 7.0 ÷ 10 or 0.70 inches. The mean effective pressure is, then, 0.7 × 80 or 56 pounds per square inch.

Suppose that the compressor cylinder has a diameter of 10 inches, length of stroke 24 inches, and speed 80 revolutions per minute.

The area of the piston is $0.785 \times 10^2 = 78.5$ square inches, and the length of the stroke is $24 \div 12 = 2$ feet.

The indicated horsepower $\dfrac{p \times l \times a \times n}{33{,}000}$

$$\frac{56 \times 2 \times 78.5 \times 80}{33{,}000} = 21.3$$

horsepower

To find the indicated horsepower of the steam engine which drives the compressor, it is necessary to proceed exactly in the same manner as for obtaining the horsepower of the compressor. The indicated horsepower of the steam engine will always be larger than the indicated horsepower obtained from the compressor. This is due to the fact that in transferring energy from the engine to the compressor there is a loss, as the friction of the moving parts of the engine and the compressor must be overcome. The ratio of the indicated horsepower of the compressor to the indicated horsepower of the steam engine is called the overall *mechanical efficiency* and is given by the following equation:

$$\text{Mechanical efficiency} = \frac{\text{indicated horsepower of compressor}}{\text{indicated horsepower of engine}}.$$

Dimensions and Capacity of Compressor and Engine.—Because the indicated horsepower developed by the steam engine is greater than that developed by the compressor, the size of the steam engine is somewhat larger than the size of the compressor. Also, it often happens that the ammonia vapor is compressed to a higher pressure than the steam pressure at the throttle of the engine. This would naturally require a smaller compressor cylinder.

Referring to the table (p. 310) showing the dimensions of compressors, it will be found that the diameter of the compressor cylinder is usually slightly less than the diameter of the engine cylinder; and the stroke of the compressor is much less than the stroke of the engine. In large machines, the speed of the compressor will be the same as that of the engine, for the compressor is directly connected with the engine. The mean effective pressure of the compressor can be about the same as or even greater than the mean effective pressure of the engine, but the product

of the piston area and the length of stroke of the compressor will be less than those of the engine; for, as has been shown, the indicated horsepower of the compressor is less than the indicated horsepower of the steam engine.

Refrigerating machines of the horizontal type have the engine connected to one end of the shaft and the other end connected to the compressor. A flywheel is placed between the two cranks. Because the stroke of the compressor is less than the stroke of the engine, the crank of the compressor is made shorter than the crank of the engine. There is an unfavorable distribution of the pressures in the two cylinders. The steam engine has a maximum pressure which occurs at the beginning of the stroke, whereas the compressor has a maximum pressure at the end of the stroke. This is unfavorable, since the maximum steam pressure occurs at a time when it is least needed. In order partly to overcome this difficulty, the cranks of the compressor and of the engine are generally set 90 degrees apart. Another way to produce the same result is to place the compressor cylinder in a vertical position and the steam-engine cylinder in a horizontal position. In this case, both compressor and engine act on the same crank. A flywheel is added to produce a smoothly running machine. The former stores up energy when the steam pressure is highest and returns it again when the pressure in the compressor is greatest.

TABLE XII.—DIMENSIONS AND CAPACITIES OF AMMONIA COMPRESSORS AND STEAM ENGINES

Ammonia compressor (double-acting)				Steam engine		
Refrigerating capacity, tons in 24 hours	Diameter of cylinder, inches	Stroke, inches	Horsepower required	Revolutions per minute	Diameter of cylinder, inches	Stroke, inches
5	$5\frac{7}{8}$	$12\frac{5}{8}$	10	75	9	14
15	$8\frac{1}{4}$	15	23	70	11	16
20	$9\frac{1}{2}$	15	30	70	10	30
50	$12\frac{13}{16}$	$21\frac{1}{4}$	63	66	16	36
75	14	30	94	60	18	42
100	$18\frac{1}{2}$	$27\frac{9}{16}$	125	50	22	42

Testing Refrigerating Plants.—It is sometimes desired to test a refrigerating plant in order to determine its capacity or efficiency under certain operating conditions. It is, therefore, useful to know how to make such tests and how to work out results from the data or readings obtained during the test. In order to show how this is done, data are given from a typical test of a refrigerating plant with a double-acting compressor.

The following is a brief description of the refrigerating system and of the conditions under which the tests were made: they were made with a double-acting, horizontal compressor directly connected to a horizontal Corliss engine, the plant having a rated refrigerating capacity of 15 tons in 24 hours. The steam cylinder was 9 inches in diameter and had a 24-inch stroke, and the piston rod was 1.68 inches in diameter. The ammonia cylinder was 8 inches in diameter and had a 16-inch stroke. The diameter of the piston rod was 1.68 inches. The ammonia condenser had two vertical tiers of 8 pipes each and was 17 feet in length. The condenser was a double-pipe type, the outer pipe being 2 and the inner pipe being 1¼ inches in diameter. The brine cooler was made of double pipes and had one vertical tier of 15 pipes 17 feet long. The inner or brine pipe was 2, whereas the outer pipe was 3 inches in diameter. The cooling surface was approximately 158 square feet. The ammonia leaving the compressor passed to the oil trap, then to the condenser, and from the condenser to the liquid receiver; from the liquid receiver, it passed to the brine cooler and again returned to the compressor. Between the liquid receiver and the brine cooler, there were installed two weighing drums for the purpose of weighing the ammonia, the latter being discharged from one drum during the time that the second drum was being filled. These drums rested upon scales, so that the ammonia could be weighed, and valves were provided, so that the drums could be alternately filled with ammonia.

The brine was pumped through the brine cooler by means of a centrifugal pump provided with a bypass, so that the quantity of brine passing through could be regulated. From the brine cooler, it passed through a double-pipe reheater and, passing through the inner pipe, was heated by steam introduced between the inner and the outer pipes. By regulating the amount of steam, the temperature at which the brine entered the brine cooler could be kept constant. From the reheater, the brine passed

to a tank from which it was allowed to flow into two measuring tanks. These were graduated to pounds at a temperature of 70° F. and for brine having 25 per cent calcium chloride. From these tanks, the brine was again pumped through the brine cooler by means of the brine pump, so that the operation was continuous. The condensing water was taken from the mains and, after passing through the ammonia condenser, was allowed to flow into two graduated tanks, one of which could be filled while the other was being emptied, so that the weight of the cooling water could be determined. During the time of the tests, indicator diagrams were taken from the steam cylinder and also, from the compressor at 10-minute intervals. The exhaust from the engine was condensed and weighed. Pressures of ammonia and of steam were taken at regular intervals by means of gages which had previously been tested. Thermometer wells were inserted in the different pipes for the purpose of determining the temperature of the ammonia, brine, and cooling water. Knowing the differences of temperature and the amount of brine, ammonia, and water used per hour, the transfer of heat in the different parts of the system could be determined.

Method of Making the Tests.—The tests were two in number, a different back pressure being used in each. Each test was made in the following manner: the engine and compressor were started and allowed to operate at the pressure decided upon until the conditions became constant. After all temperatures became constant, the test was started. All temperature readings and indicator cards from engine and compressor were taken at 10-minute intervals. The ammonia, brine, condensing water, and condensed steam were collected and weighed on scales or measured in graduated tanks, as before described. The tests proper were of from 1 to 2 hours' duration.

TEST OF DOUBLE-ACTING 15-TON REFRIGERATING PLANT

Item		
1.	Duration of test, hours	2
2.	Suction pressure (by gage), pounds per square inch	21
3.	Condenser pressure (by gage), pounds per square inch	117.1
4.	Revolutions per minute	70.7
5.	Temperature of brine, inlet, degrees Fahrenheit	47.0
6.	Temperature of brine, outlet, degrees Fahrenheit	17.9
7.	Difference, inlet and outlet temperature, degrees Fahrenheit	29.1
8.	Specific heat of brine, degrees Fahrenheit	0.757
9.	Weight of brine circulated, pounds	16,300
10.	Weight of brine circulated per hour, pounds	8,150
11.	Cold produced, B.t.u. per hour	179,533
12.	Tons capacity in 24 hours	14.96

TEST OF DOUBLE-ACTING 15-TON REFRIGERATING PLANT.—(*Continued*)

Item			
13.	Temperatures of condensing water, degrees Fahrenheit.	Outlet to condenser	66.75
14.		Inlet to condenser	52.00
15.		Difference, outlet and inlet	14.75
16.	Weight of cooling water used, pounds		28,630
17.	Weight of cooling water used per hour, pounds		14,315
18.	B.t.u. absorbed per hour by cooling water		211,146
19.	Temperatures of ammonia, degrees Fahrenheit.	At condenser inlet	71.9
20.		At condenser outlet	54.5
21.		Difference, inlet and outlet of condenser	17.4
22.		At cooler inlet	61.1
23.		At cooler outlet	7.8
24.		Difference cooler inlet and outlet	53.3
25.	Weight of ammonia used, pounds		710.5
26.	Weight of ammonia used per hour, pounds		355.25
27.	Weight of dry steam used, pounds		1,416
28.	Weight of dry steam used per hour, pounds		708
29.	Mean effective pressures	Head end, steam cylinder, pounds per square inch	36
30.		Crank end, steam cylinder, pounds per square inch	31.5
31.		Head end, ammonia cylinder, pounds per square inch	53.7
32.		Crank end, ammonia cylinder, pounds per square inch	52.6
	Indicated horsepower	Head end, steam cylinder	9.82
34.		Crank end, steam cylinder	8.29
35.		Total, steam cylinder	18.11
36.		Head end, ammonia cylinder	7.72
37		Crank end, ammonia cylinder	7.22
38.		Total, ammonia cylinder	14.94
39.	Mechanical efficiency, per cent		82.5
40.	Weight of dry steam per indicated horse power per hour, steam cylinder, pounds		39.1
41.	Weight of dry steam per hour per ton of refrigeration in 24 hours, pounds		47.3

METHOD OF CALCULATING TEST

Item

$$7 = \text{item } 5 - \text{item } 6 = 47.0 - 17.9 = 29.1$$

$$10 = \text{item } 9 \div \text{item } 1 = 16{,}300 \div 2 = 8{,}150$$

$$11 = \text{item } 10 \times \text{item } 8 \times \text{item } 7 = 8{,}150 \times 0.757 \times 29.1 = 179{,}533$$

$$12 = \frac{\text{item } 11 \times 24}{288{,}000} = \frac{179{,}533 \times 24}{288{,}000} = 14.96$$

$$15 = \text{item } 13 - \text{item } 14 = 66.75 - 52.00 = 14.75$$

$$17 = \text{item } 16 \div \text{item } 1 = 28{,}630 \div 2 = 14{,}315$$

$$18 = \text{item } 17 \times \text{item } 15 = 14{,}315 \times 14.75 = 211{,}146$$

$$21 = \text{item } 19 - \text{item } 20 = 71.9 - 54.5 = 17.4$$

METHOD OF CALCULATING TEST.—(*Continued*)

Item

24 $= \text{item } 22 - \text{item } 23$
$= 61.1 - 7.8 = 53.3$

26 $= \text{item } 25 \div \text{item } 1$
$= 710.50 \div 2 = 355.25$

28 $= \text{item } 27 \div \text{item } 1$
$= 1{,}416 \div 2 = 708$

33 $$= \frac{\text{item } 29 \times 2 \times 63.62^* \times \text{item } 4}{33{,}000}$$
$$36 \times 2 \times 63.62 \times 70.7 =$$

34 $$= \frac{\text{item } 30 \times 2 \times 61.4^* \times \text{item } 4}{33{,}000}$$
$$= \frac{31.5 \times 2 \times 61.4 \times 70.7}{33{,}000}$$

35 $= \text{item } 33 + \text{item } 34$
$= 9.82 + 8.29 = 18.11$

36 $$\frac{\text{item } 31 \times 16 \times 50.27^* \times \text{item } 4}{33{,}000 \times 12}$$
$$= \frac{53.7 \times 16 \times 50.27 \times 70.7}{33{,}000 \times 12} = 7.72$$

37 $$= \frac{\text{item } 32 \times 16 \times 48.05^* \times \text{item } 4}{33{,}000 \times 12}$$
$$= \frac{52.6 \times 16 \times 48.05 \times 70.7}{33{,}000 \times 12} \quad 7.22$$

NOTE.—In practice, the indicated horsepower would be calculated from each indicator diagram separately and the average of the indicated horsepowers taken, instead of using the average mean effective pressure and average revolutions per minute, as is done here.

38 $= \text{item } 36 + \text{item } 37$
$= 7.72 + 7.22 = 14.94$

39 $= \text{item } 38 \div \text{item } 35$
$= 14.94 \div 18.11 = .825$

40 $= \text{item } 28 \div \text{item } 35$
$= 708 \div 18.11 = 39.4$

41 $= \text{item } 28 \div \text{item } 12$
$= 708 \div 14.96 = 47.3$

Heat Balance.—The ammonia travels through a complete circuit, taking up heat at some points and giving it out at others. These amounts should balance, since the ammonia always returns to the condition in which it started. The heat balance serves to show this balance and also tells where and in what quantities the heat is transferred.

* Effective area of piston in square inches.

The method of calculating the quantities given in the heat balance follows:

Heat Discharged from Compressor.—Ammonia at the rate of 355.25 pounds per hour enters the compressor cylinder at a gage pressure of 21 pounds per square inch and at 7.8° F.; 21 pounds per square inch gage pressure corresponds to 21 + 14.7 or 35.7 pounds per square inch absolute pressure.

From the table of properties of saturated ammonia (p. 492), the amount of heat in 1 pound of the ammonia vapor above the heat in the liquid ammonia at −40° F. is equal to heat of the liquid + latent heat + superheat = 614. B.t.u. per pound.

Work of compression per hour

$$= 14.94 \times 2{,}545 = 38{,}022 \text{ B.t.u.}$$

$$\frac{38.022}{355.25} = 107 \text{ B.t.u. per pound of ammonia.}$$

NOTE.—14.94 is the indicated horsepower of the ammonia cylinder and 2,545 is the number of B.t.u. equivalent to the indicated horsepower acting for 1 hour.

Heat in 1 pound of ammonia discharged from the compressor = 614 + 107 = 721 B.t.u.

Heat in 355.25 pounds of ammonia discharged from the compressor

$$= 355.25 \times 721 = 256{,}135 \text{ B.t.u.}$$

Heat Lost between Compressor and Condenser.—The heat leaving the compressor in 1 pound of ammonia vapor is 721 B.t.u., being made up of

Heat of liquid + latent heat + superheat.

The heat of the liquid and the latent heat being taken at the discharge pressure = 117.1 + 14.7 or 131.8 pounds per square inch absolute pressure.

From the table of properties of saturated ammonia, the temperature corresponding to 131.8 pounds per square inch absolute pressure is 71.3° F.

The latent heat corresponding to 131.8 pounds per square inch absolute is 507.3 B.t.u.

The superheat is $(t - 71.3) \times 0.58$.

NOTE.—0.58 is the specific heat of ammonia vapor for the pressure and temperature given.

The heat of the liquid per pound of ammonia = 122.0 B.t.u. Therefore,

$$721 = 122 + 507.3 + (t - 71.3) \times 0.58$$

or

$$t - 71.3 = \frac{721 - 629.3}{0.58} = \frac{91.7}{0.58} = 158.1° \text{ F.}$$

and

$$t = 158.1 + 71.3 = 229.4° \text{ F.}$$

This is the temperature of the ammonia vapor leaving the compressor.

Temperature of ammonia vapor entering condenser = 71.9° F.

Heat lost per pound of ammonia between compressor and condenser = 0.58(229.4 − 71.9) = 0.58 × 157.5 = 91.4 B.t.u.

Total heat lost per hour between compressor and condenser = 355.25 × 91.4 = 32,470 B.t.u.

Heat Lost in the Condenser.—Temperature of ammonia vapor entering condenser = 71.9° F.

Total heat of ammonia vapor entering condenser is 630 B.t.u. per pound.

Temperature of ammonia liquid leaving condenser is 54.5° F.

Heat in 1 pound of ammonia leaving the condenser = 103 B.t.u.

Heat removed in condenser per pound of ammonia.

$$\begin{aligned} &= (630 - 103)355.25 \\ &= 527 \times 355.25 = 187{,}217 \text{ B.t.u.} \end{aligned}$$

NOTE.—The *heat lost between compressor and condenser* might have been calculated more easily by subtracting the *heat lost in the condenser plus the heat of the liquid of the ammonia leaving the condenser* from the *heat discharged from the compressor.* The calculations have been made in the way shown here in order to demonstrate how the temperature of the ammonia vapor leaving the compressor may be calculated.

Heat Gained between Condenser and Cooler.—Temperature of ammonia at inlet of brine cooler = 61.1° F.

Temperature of ammonia at outlet of condenser = 54.5° F.

Heat gained per pound of ammonia between the condenser and the brine cooler = 110.5 − 103 = 7.5 B.t.u.

Total gain on heat of ammonia between condenser and brine cooler = 7.5 × 355.25 = 2,664 B.t.u.

Heat Gained in Cooler.—The heat gained in the brine cooler is equal to the heat in the ammonia vapor leaving the brine cooler minus the heat in the ammonia entering the cooler.

The temperature of the liquid ammonia entering the cooler is 61.1° F. The heat of liquid in the ammonia liquid entering the brine cooler is 110.5 B.t.u. per pound.

The heat per pound of ammonia vapor leaving the brine cooler is made up of the heat of liquid, latent heat of evaporation at 35.7 pounds per square inch absolute, and the heat necessary to superheat the vapor.

The heat in the ammonia vapor per pound = 614 B.t.u.

Therefore, the heat gained per pound of ammonia in the brine cooler

$$= 614 - 110.5 = 503.5 \text{ B.t.u.}$$

Total heat gained by the ammonia in the brine cooler

$$= 503.5 \times 355.25 = 178{,}868 \text{ B.t.u.}$$

Summary of Heat Cycle

	Heat gained	Heat lost
Work of compression	38,022	....
Between compressor and condenser	....	32,470
To condensing water	....	187,217
Between condenser and cooler	2,664	....
In cooler	178,868	....
Total	219,554	219,687

The heat cycle should, of course, balance exactly. In this case, it is out of balance 219,687 − 219,554 = 133 B.t.u.

CHAPTER IX

ICE MAKING

Ice-making Systems.—The present system of mechanical refrigeration owes its origin to the necessity for manufacturing ice. Mechanical refrigeration, although first used in making ice, now finds many other applications, as in cold storage, candy factories, sugar refineries, chemical works, and marine service. Since ice was first manufactured, methods have become more or less standardized, so that the present systems and apparatus employed are very much alike.

There are two systems of ice making in general use at the present time—the can and the plate. These two systems differ from each other in the relative location of the freezing medium and the water to be frozen.

In the *can system*, the water to be frozen is contained in metal cans placed in large tanks containing cold brine which circulates around the cans. In this process, the water begins to freeze from the outer walls of the can toward the center.

In the *plate system*, the water to be frozen surrounds the large metal plates, which contain coils or cells filled with expanding ammonia or cold brine. In this system, the water first begins to freeze on the outer walls of the plates.

Transparent Ice.—Ice which is not injurious to the health when taken internally by a person is said to be wholesome and sanitary and is called *hygienic ice*. Any suitable drinking water may be used for producing hygienic ice. Ice which is not transparent may be pure and wholesome. The presence of air in the water to be frozen makes the ice opaque and non-transparent. A cake of ice frozen by the can system may have a milky appearance at its center. This is due to the presence of air in the water and is not injurious. Transparent ice may be made by the can system through the use of properly distilled water.

In order to meet the public demand for clear and crystal ice, considerable ingenuity has been exercised, and complicated methods have been used, which, of course, add to the cost of production.

Analysis of Water.—Pure water is composed of hydrogen and oxygen, without even a trace of any other substance. As water forms, it seizes hold of various gases in the atmosphere. In this way, carbonic acid, ammonia, and nitrates are absorbed. Rain containing these chemicals falls to the earth where it reacts and forms different salts. A water containing large quantities of salts is said to be *hard,* and one containing few salts is said to be *soft.* To insure clear ice, an analysis of the water should be made, and if it shows large amounts of salts, it should be softened by chemical and mechanical means.

In some cases it is necessary to make a sanitary and mineral analysis of the water that is to be used for an ice plant. The sanitary analysis is a bacteriological examination and determines whether or not the water is suitable for human consumption. This type of analysis is generally made by the local or state board of health. The mineral analysis determines the amounts of the dissolved minerals found in the water. In Table XII*a* is shown the form used in reporting the water analysis. The suspended matter consists of dirt, sand, and clay, that is, in suspension;

TABLE XII*a*.—FORM OF WATER ANALYSIS REPORT

PHYSICAL CHARACTERISTICS	Parts per Million	Grains per Gallon
Suspended matter.....		
Total dissolved solids..		
Turbidity............		
Color.		
Odor..		

CHEMICAL CHARACTERISTICS (Expressed in terms of calcium carbonate)

	Parts per Million	Grains per Gallon
Hardness.............		
Alkalinity (P).........		
Alkalinity (M).........		
Mineral acidity........		

(NOTE.—P indicates reaction to phenolphthalein and M indicates reaction to methyl orange.)

CHEMICAL COMPOSITION

INCRUSTING SOLIDS	Parts per Million	Grains per Gallon
Calcium carbonate.....		
Calcium sulphate......		
Calcium chloride......		
Magnesium carbonate..		
Magnesium sulphate...		
Magnesium chloride...		
Iron oxide...........		
Aluminum oxide.......		
Silica................		
Suspended matter.....		

NON-INCRUSTING SOLIDS	Parts per Million	Grains per Gallon
Sodium sulphate.......		
Sodium chloride.......		
Sodium carbonate......		
Dissolved Gases:		
Free carbon dioxide....		
Hydrogen sulphide.....		

(NOTE.—To convert parts per million to grains per gallon multiply by 0.06 or to convert grains per gallon to parts per million multiply by 17.12.)

the total solids representing the dissolved mineral matter. Its hardness is a measure of all the calcium and magnesium compounds in the water; and this is commonly expressed for con-

TABLE XIIb.—EFFECT ON ICE OF MINERALS

Minerals in water	Effect in ice	Result of treatment with hydrated lime
Calcium carbonate	Forms gritty, dirty, discolored deposit, usually in lower part and center of the cake. Causes shattering at low freezing temperatures	Practically eliminated
Magnesium carbonate	Forms gritty, dirty, discolored deposit, milky patches and bubbles, and also causes shattering at low freezing temperatures	Practically eliminated
Iron oxide	Causes bad discoloration, yellow or brown deposits and also stains the calcium and magnesium deposits	Eliminated
Aluminum oxide and silica	Cause dirty deposit and sediment	Practically eliminated
Suspended matter	Causes dirty deposit and sediment	Eliminated
Calcium sulphate. Calcium chloride	Act like and are no worse than sodium sulphate and sodium chloride. Do not form deposit	No change
Magnesium sulphate. Magnesium chloride	Cause greenish or grayish cast, concentrate in core water, retard freezing and cause heavy cores. Often show up as white spots and dirty colored streaks. Also act like sodium sulphate and sodium chloride. Do not form deposit	Changes to calcium sulphate. Changes to calcium chloride
Sodium sulphate. Sodium chloride	Cause white butts, concentrate in core, make heavy core and retard freezing—but do not form deposit	No change
Sodium carbonate (actually present as sodium bicarbonate)	In only small quantities often causes shattering at temperatures below 16°. Also causes white butts, concentrates in core, retards freezing and makes heavy cores. Does not form deposit	Treatment changes sodium bicarbonate to sodium carbonate—treatment improves but little

venience in terms of calcium carbonate. The alkalinity can be determined by the use of indicator phenolphthalein, but this test is not necessary except in treated water. In the chart this is represented by alkalinity (*P*). The alkalinity (*M*) is a test

made by using methyl orange as an indicator, which measures all the carbonates, bicarbonates, and hydrates in the water, which is also expressed for convenience in terms of calcium carbonate. The mineral acidity is a measure of the acid characteristic of the water, which is rarely found except in mine and waste water. The chemical composition shows the amounts of incrusting and non-incrusting solids.

The refrigerating engineer is interested in the effect that the various constituents of water will have on the quality of ice. These effects are clearly shown in Table XII*b*. Magnesium sulphate and calcium chloride are usually present with the carbonate of calcium and magnesium and, therefore, require lime treatment which changes them to the less objectionable ingredients calcium sulphate or calcium chloride. The amounts of calcium and magnesium carbonates, and of oxide of iron, the suspended matter, and the color of the water determine whether or not water treatment is necessary.

The following is an analysis of a hard water, which shows the amounts of salts in solution:

Calcium carbonate..........	34.4092 grains per U. S. gallon
Magnesium chloride.........	16.3755 grains per U. S. gallon
Magnesium sulphate........	7.2917 grains per U. S. gallon

This water is heavily impregnated and is a very difficult water to handle for "raw-" or natural-water ice making.

Defects of Ice.—It frequently happens that the ice produced is not perfectly clear, has a white core, or even some taste or flavor.

Milky Ice.—Ice that has a milky appearance is generally the result of small air bubbles in the distilled water, due to insufficient boiling in the reboiler. It may be the result of too rapid condensation in the condenser causing more air to be drawn in than can be removed by the reboiler. This can be prevented by reducing the quantity of condensing water, thereby increasing the pressure of the steam in the boiler. Air frequently leaks into the distilled water pipe or may get into the water during the process of can filling.

White-core Ice.—In plants using distilled water, this is sometimes caused by overworking the boiler. There is an accumulation of mineral matter in the boiler water, often due to the fact that the boiler has not been cleaned so often as it should be.

Carrying too much water in the boiler and lack of attention to "blowing off" will also produce this defect. More often, though, the white core is the result of the carbonates of lime or magnesia in the water. As the water in the cans begins to freeze to the walls, these carbonates are rejected to the unfrozen water. Since the center is the last to freeze, this water becomes saturated with these carbonates, thus causing the white core.

Red-core Ice.—If manufactured ice is found to have a red core, it indicates the presence of carbonate of iron from which oxide of iron has been separated. The oxide of iron nearly always comes from the iron pipes and coils in the plant. In order to prevent this defect, the pipes when idle should be kept filled with water which has been distilled and reboiled.

Rotten Ice.—Cakes of ice which are hollow in the center or are incomplete otherwise are said to be "rotten." This condition increases the surface exposed to the air which causes it to melt rapidly. Great care should, therefore, be taken to have no holes in the ice cakes and to insure that they are solidly frozen.

Purifying Water for Ice Making.—In the modern refrigerating plants using raw water for making ice, it is necessary, in practically all cases, to remove the organic matter, iron, clay, and sand, which may be in suspension in the water. For this purpose, special types of water filters have been designed. The filtering material is usually alternate layers of coarse and fine sand, crushed flint quartz being usually preferred. Such filtering devices are usually supplemented by charcoal filters provided to remove coloring matter, objectionable odors, and tastes. The filters can be used successfully for removing the organic matter and minerals held in suspension. Occasionally, however, a kind of water that has iron in *solution* is to be used in refrigerating plants; in this unusual circumstance, a special device must be designed which will pass compressed air in minute bubbles through the water. Mineral matter held in *solution* in water, making it *hard*, is ordinarily removed by processes included under the term *water softening*. There are no general rules to be laid down for the softening of natural or raw water, and in every plant the chemical treatment must be determined from a mechanical analysis of the water. Alum is often satisfactorily used, and some types of water filters are provided with a so-called *alum pot*, which is connected into the pipe supplying the raw water to the filter apparatus. Such an alum pot is shown in Fig. 207.

Operation of Alum Pot.—When alum is introduced into water with an alkaline reaction, a flaky precipitate (aluminum hydroxide) is formed, which is insoluble in water. This flaky precipitate binds together into bits of gravel the very fine particles of suspended organic and mineral matter which would otherwise pass through the sand. These bits of gravel collect on top of the sand in the filter, where they add to the deposit of filtering material. It is usually recommended that the operators of ice-making plants

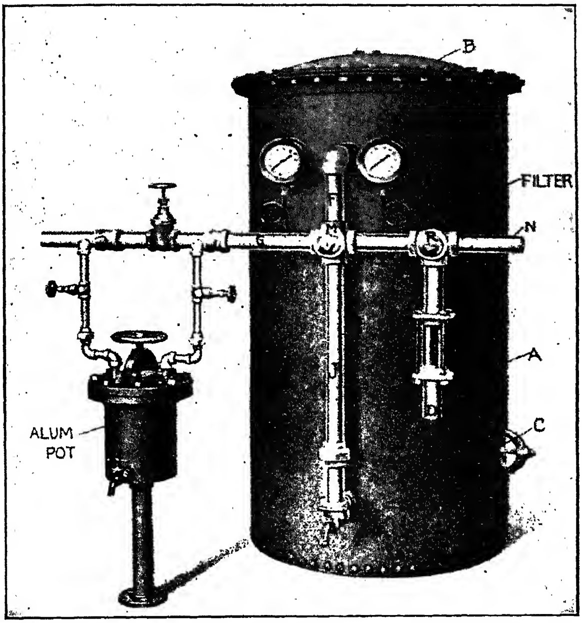


FIG. 207.—Pressure water filter and alum pot.

determine by trial how much alum is to be introduced into the water for softening. If too much alum is used, it has the same effect on the ice as any other soluble material. For making the core of the ice clear, best results are obtained by reducing the amount of alum introduced until the clearest core is obtained. Rock alum and not the powdered kind should be used, and the pot should be inspected frequently to see that it is kept well filled.

Pressure Type of Water Filter.—A typical water filter of the pressure type is shown in Fig. 207. It consists of a cylindrical steel shell *A* of which the flanged top can be removed for cleaning

and repairing. There is also a hand hole at C near the bottom, to make it easy to remove the beds of filtered material. The raw water to be filtered enters near the top through the pipe F. The end of this pipe inside the shell is fitted with a galvanized distributing funnel located centrally over the filtering material. There is a branched outlet pipe in the bottom of the filter, and this is provided with so-called *umbrella nozzles*, designed to give a good distribution to the *wash water* through the filtering sand.

In the operation of this filtering device, water flows down through the beds of filtering sand and discharges from the bottom into the pipe J, which extends upward so that the filtered water passes through the four-way valve into the pipe N, shown extending to the right in the figure.

Pressure gages are placed in the water inlet and discharge pipes of the filter, to show the loss of pressure of the water in passing through the filtering beds. When this loss of pressure becomes excessive, it is an indication that the filter needs cleaning. The permissible pressure drop through the filter depends, of course, on the initial water pressure. For example, if the gage pressure of the water in the inlet pipe is 60 pounds per square inch, a pressure drop through the filter of 5 pounds per square inch is about average practice; and when the pressure drop becomes as much as 10 pounds per square inch, the filter needs cleaning. The filtering beds should not be washed oftener than is absolutely necessary, because the material which collects on top of the bed increases its efficiency. In order to wash the filtering sand, the four-way cock M should be turned so that the water will flow from the inlet pipe G downward through the pipe J, then upward through the layers of filtering sand inside the filter and discharge through the pipe N. The water used for washing cannot be used for ice making, so that it should be discharged into the drain-pipe D. A sight glass is provided in a section of the pipe D, so that a person engaged in washing the filter may observe, from the appearance of the water discharged into the drain, when the washing has been carried far enough.

Distilled-water System.—If so-called *raw* or natural water, as taken from wells, rivers, or ponds is frozen by artificial means without being distilled before freezing and without agitation while freezing, the ice produced will have a whitish, marble-like appearance due to the air which is always present in natural water. This kind of ice is perfectly good, as far as any useful

purpose is concerned, but most people, at least in America, do not like to use it for table or household purposes.[1]

The distilled-water system was one of the first successful devices for making clear artificial ice. Briefly, in this system, the exhaust steam from the steam engine is condensed at about atmospheric pressure in a condenser and then passes on to a reboiler, where the water is heated to a temperature at which it will boil again while exposed to the atmosphere, to permit the air in the condensed steam to be separated out and pass off from the surface of the liquid. This reboiler should have a skimmer or surface blowoff to remove any foreign matter which may accumulate on the surface of the condensed steam, especially the oil used for lubricating the engine. After the condensed steam is drawn off from the reboiler, it is passed through a set of water-cooled coils, where it is cooled to about 70 to 80° F., after which it goes through a sand filter and then a charcoal filter, the latter being especially useful in removing the last traces of any oil used in the engine. After being discharged from the two filters, the distilled water is ready to be frozen into cakes of ice, unless some system of precooling is used in the plant. The ice made from this distilled and filtered water will be entirely transparent if care is taken not to let air get into the water while it is freezing. Figure 208 shows the arrangement of water-distilling apparatus in an ice-making plant. As arranged here, the exhaust steam, after leaving the engines and pumps, passes through a feed-water heater, losing some of its heat and, at the same time, heating the boiler feed water. From the feed-water heater, the exhaust steam passes to an oil trap, where a large part of the oil is removed. It then passes to the condenser, where it is condensed by being brought in contact with water-cooled surfaces. The water, upon leaving the condenser, passes to a reboiler, where it is slowly boiled in order to remove the air. The rate of flow through the reboiler is such that any oil which may be present will rise to the surface and be removed there by a skimming device. Any gases passing off are collected under a hood from which they escape to the atmosphere. From the reboiler, this purified water is passed to a

[1] Transparency in ice is not a requirement when the ice is to be used as the cooling medium in railroad refrigerator cars, in packing fish for transportation, and in ice-cream packing. For these purposes, it is unnecessary to go to the expense of either distilling the water used in making ice or providing mechanical means for removing the air in the water. At present, there are few distilled-water ice plants being installed.

forecooler, where its temperature is lowered by the removal of heat. The heat is removed by coils containing circulating cold water. The economy of the plant may be increased if the boiler feed water is utilized to perform this cooling before it reaches the feed-water heater.

After the water is filtered, it is chilled by brine or direct-expansion coils before being used to fill the ice cans. Before the chilled water enters the cans, it is usually passed through sponge filters to insure absolute purity.

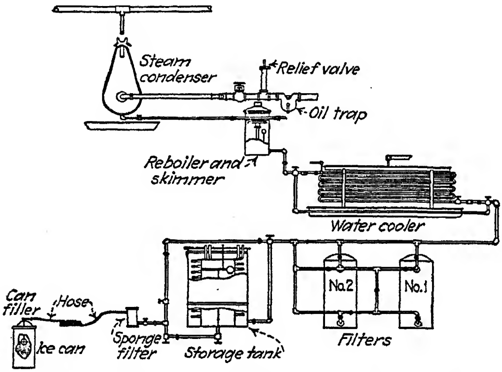


FIG. 208.—Distillation equipment for ice-making plant.

Where Distilled-water Ice-making Plants Must Be Used.—In some places, it is necessary to use a system of making ice from distilled water. Such places are usually where the natural water contains a large amount of mineral matter which cannot be readily precipitated—for example, sodium salts. In such places, there is still the possibility of using electric motors or oil engines for motive power by the application of a series of evaporating tanks[1] in case the cost of fuel and operation of a steam plant are excessive. For any set of conditions, it is necessary to work out the best method of removing insoluble mineral matter from the water, and cases of this kind can be decided only after careful study of all the conditions. While it is true that there are very few new plants now being equipped to use distilled water for ice making, one must realize that there are still a great many distilled-water plants in operation. Every year, however, there are a number

[1] See MACINTIRE, "Handbook of Mechanical Refrigeration," p. 378.

of distilled-water steam-operated plants changing over to electric operation. One of the serious objections to the distilled-water system of ice making is that the distilling apparatus deteriorates very rapidly.

The Plate Raw-water System.—When raw or natural water was first used in ice-making plants, the ice was frozen in large slabs or plates. In this system, the refrigerant is circulated in expansion coils between large, flat metal plates so as to maintain the plate at a temperature of about 0° F. or lower while it is submerged in a tank of water which is to be frozen. The ice is formed on one side only on this plate, and the water near the freezing surface is kept in constant agitation. By this method, after somewhere between 5 and 7 days of freezing, a slab of ice will be formed on the refrigerated metal plate, the thickness of the ice slab being between 10 and 12 inches. When the required thickness of ice has been formed, the refrigerant is turned off from the coil adjacent to the plate on which the ice is being formed, and then hot ammonia vapor (in an ammonia plant) is discharged into the coil. This hot vapor melts the slab of ice from the metal plate to which it was attached. After the slab of ice is detached, it is lifted from the ice tank by a crane. Such a slab of ice usually weighs between 3 and 5 tons. After it is lifted by the crane from the freezing tank, it is carried away to a suitable table, where it is sawed into cakes of ice.

By the plate method, a very good quality of ice can be made. It has, however, the disadvantage that the cakes of ice are not uniform in thickness, and this fact causes some objection from dealers. It must also be noted that the freezing tank for the plate system must be made deeper than the tank used for other systems of ice making, and, because of this greater depth, the headroom over the ice tank for the operation of the crane must also be greater. Another item of expense is the cost of power to operate the saw on the table where the slab of ice is cut up into cakes. Briefly, the disadvantages of the plate system as compared with the systems more generally used are (1) that the first cost is greater, (2) that there is trade resistance because of the non-uniform thickness of the cakes, and (3) that there is greater expense for operating the plant. Because of these items of greater initial costs, operating expenses, and trade resistance, no ice-making plants are now being installed that are equipped for making ice in slabs or plates.

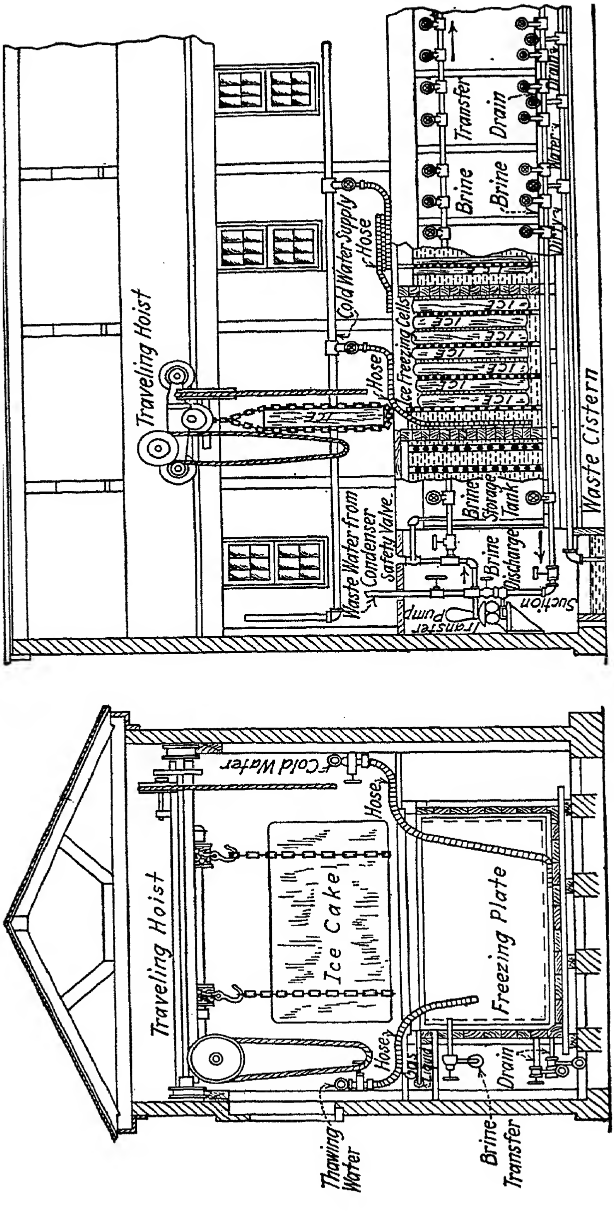


FIG. 209.—Equipment for making plate ice.

A general arrangement of the freezing tanks in a plate ice-making plant is shown in Fig. 209. The plates shown here are made up with freezing coils arranged so that brine may be circulated through them. An auxiliary brine pump is used to circulate the brine through the freezing coils. In order to prevent the pump from subjecting the freezing coils to undue pressure, a safety valve connected to the discharge of the pump conveys the brine back into the storage tank, when a pressure exceeding that of the setting of the safety valve occurs.

As shown in the figure, the cakes of ice are removed from the plates by means of warm water, which is allowed to pass either directly from a hose into the ice-freezing cells or into the freezing coils, melting the cakes free from the plates. The circulating water from the condenser furnishes warm water at a temperature sufficient to perform this operation.

The Can Raw-water System.—The method of making artificial ice in cans is very old; in fact, almost innumerable variations have been advocated and also practically applied for making ice in this way. The large problem to be met in any system of making artificial ice is, of course, the elimination in the water of air, which has the effect of making the ice white and marble-like in appearance. In some of the early plants for making can ice, paddles were provided in each can for the purpose of keeping the water in agitation and thus removing the air. The method has been tried of attaching short shafts to the sides of the can, so that it could be supported on bearings and rocked by a suitable mechanical device to keep the water in agitation. Methods have also been tried of avoiding any method of agitation by special treatment of the water. In one of these latest systems, the raw water for ice making is collected in a water-storage tank, and after chemical treatment and filtering it is heated to about 150° F. in a very high vacuum.

It is a general present practice to put raw water for ice making into the cans, which are then lowered into the freezing tank where cold brine circulates around them. The level of the brine is usually about the same as the level of the surface of water inside the cans, although there is an advantage in having the level of the former slightly above the level of the latter in order to shorten the time required for freezing. The temperature of the brine for making ice in cans is uaually about 14 to 18° F. It is maintained either by means of direct-expansion piping in the

freezing tank, as in Figs. 210 and 212, or by means of a shell-and-tube cooler, Fig. 213, which is submerged in the brine in

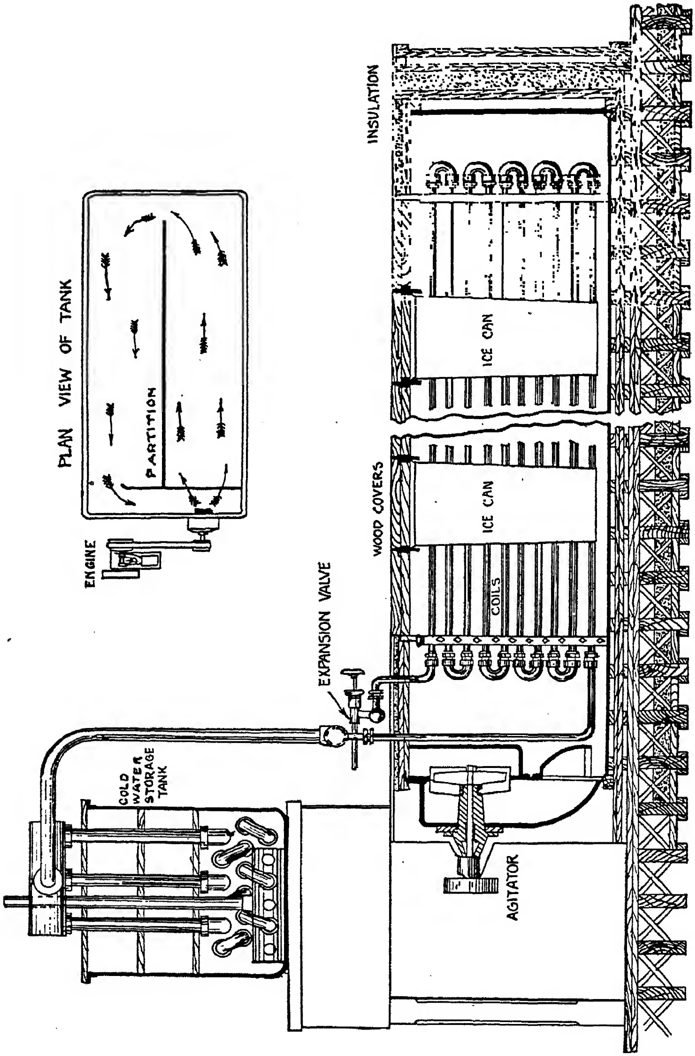


Fig. 210.—Equipment for making can ice.

the tank. In either case, the brine is kept in circulation by means of a suitable agitator driven by an electric motor. The horizontal

type, shown in Fig. 214, is used in the freezing tank in Fig. 210. The vertical type of agitator, shown in Fig. 215, is more generally used, as shown in Fig. 212.

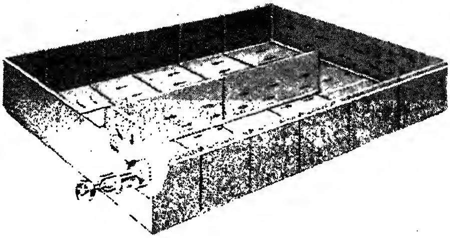


FIG. 211.—Partitions in ice-making tank.

In Fig. 210 is shown a general layout of a can system. In studying this figure, note that the cooling coils of the evaporator are submerged in the brine. An expansion valve is placed in the pipe line supplying the coils in order to regulate the flow of ammonia and thus control the temperature of the brine. At the center is shown the agitator. In Fig. 211 is shown the method

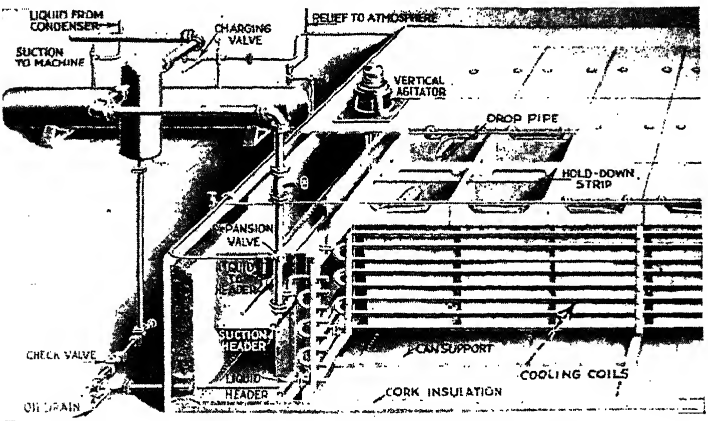


FIG. 212.—Ice-making plant equipped with vertical agitator.

of partitioning off the tank in order to improve the circulation of the brine over its entire width. When the water in the cans is frozen, they are removed from the brine by an overhead crane. They should be wiped off or allowed to drip so as to prevent brine

from entering the other cans. Then the traveling crane carries them to the thawing apparatus. In one type of thawing apparatus, the cans containing ice are brought to a horizontal position, and then warm water is sprayed on them, freeing the ice cakes

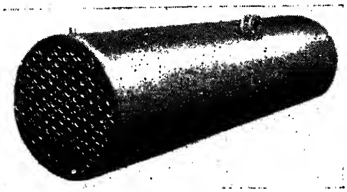


FIG. 213.—Shell-and-tube cooler.

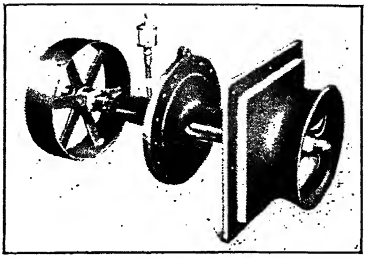


FIG. 214.—Horizontal brine agitator.

from them. Another method is to immerse the cans containing the cakes of ice in a *dipping tank* containing water warm enough to detach the ice cakes. The ice cakes, after removal from the cans, are placed on a chute which discharges them into the ice-storage room.

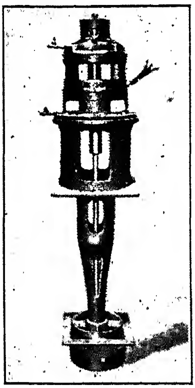


FIG. 215.—Vertical brine agitator.

In one system of making ice, the ice cans are always stationary in small freezing tanks as in Fig. 228, and have heavily insulated bottoms. The ice cakes are thawed from the sides of the cans in a freezing tank by the method of circulating around the cans brine which has been warmed by being pumped through the water storage tanks, instead of being circulated through the other freezing tanks in the plant. This use of stationary ice cans and warm brine circulation for thawing is called the *Arctic Pownall* system.

York Vertical Trunk System.—The vertical-trunk freezing system consists of an evaporating surface placed in a steel casing through which brine is circulated at a high velocity. The evaporating coil is of the herringbone type as shown in Fig. 216, and has a liquid feed header at the bottom and a suction gas header at the top. The pipes connecting these headers are of short length, shaped like a V and welded in place. This permits the entering and leaving refrigerant to flow always in

the same direction, thereby reducing the friction of flow which permits a higher rate of circulation. The coils are flooded (p. 65), and an accumulator (p. 63) or trap is provided as an integral part of the evaporating coil. The liquid refrigerant is supplied to the accumulator by means of a float valve. Any oil that reaches the evaporator may be drained from an oil leg which is connected

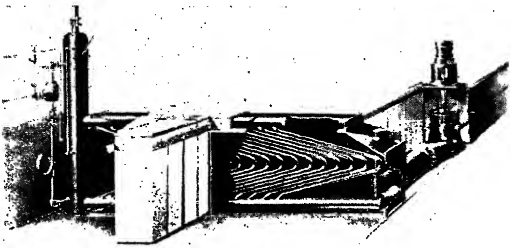


FIG. 216.—York vertical trunk system of ice-making.

to each evaporating coil. It is claimed by the manufacturer of this type of freezing coil that it has the advantages of the shell-and-tube-brine freezing system. As the brine is outside of the coils, the possibility of freeze-up or pipe splitting which has occurred with shell-and-tube coolers is eliminated. The con-

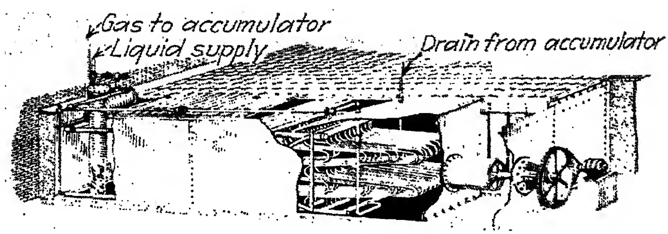


FIG. 217.—Frick verti-flow race system of ice-making.

stant-flooded condition of the evaporating coils, due to the float-valve regulation, insures a high rate of heat transfer. The heat transfer for such a system is about 100 B.t.u. per square foot per hour per degree Fahrenheit. Because of this high heat-transfer coefficient, the equivalent surface in linear feet of pipe is about 45. The brine velocity through the

"trunk" of the system is about 150 feet per minute while in the tank the velocity is about 25 feet per minute.

Frick Verti-flow Race System.—The verti-flow unit is shown in Fig. 217. It consists of five lengthwise headers with parallel short headers welded crosswise at the top and bottom. The vertical coils which are W shaped are welded to the headers. The evaporating coils are flooded thus obtaining a high rate of heat transfer. The refrigerant is controlled by a float valve. The verti-flow unit is placed in a "race" through which brine flows at a high velocity causing a large amount of heat to be transferred, thus reducing the actual amount of evaporating surface. This system has the advantages of the shell-and-tube brine cooler (Fig. 213) and is similar in principle to the York trunk system.

Air Agitation in Ice Cans.—One of the first systems used for agitating the water in ice cans for the removal of air required the use of a *drop pipe* into the center of each can about three-fourths of the way to the bottom of the can. Compressed air at a gage pressure of about 3 pounds per square inch was allowed to pass into this pipe and discharge from its lower end into the water. Because of the low pressure of the air supplied this method of obtaining air removal is often called *low-pressure air agitation.* After a certain amount of freezing, the drop pipe was removed, and the freezing was continued without any agitation of the water. There is also another method by which the drop pipe is left in the core of the cake of ice somewhat longer than in the preceding method, and then, when it is removed, there is provision for adding distilled water into the space from which the drop pipe was removed. In either of these systems, however, it is necessary to remove the drop pipe at a somewhat definite time, as, otherwise, it will be frozen into the cake of ice, and, at any rate, the usefulness of the drop pipe ceases when it begins to freeze into the ice, because the pressure of the compressed air in such a system is not great enough to prevent the formation of ice at the end of the pipe. This ice formation at the end of the pipe, of course, closes it to the further distribution of compressed air.

The quantity of low-pressure air supplied at 3 pounds per square inch gage pressure is about 0.5 cubic foot per minute per 300-pound cake of ice.

A can system of making artificial ice from raw water, using compressed air for agitation at about the pressure stated in the

last paragraph, will give satisfactory ice for marketing when fairly good natural water is used. In this method, there is, of course, always the expense of handling the drop pipes which must be removed from the cans at the proper time or from the cakes of ice by some method of thawing.

One type of drop pipe for low-pressure agitation is placed in the center of the can and is intended to be frozen into the center of the cake of ice. In this method, the drop pipe is perforated with small holes at several places along its length to permit the escape of compressed air for agitating the water at the core of the cake of ice, especially after the end of the tube has been filled up by the ice freezing at its end. The present tendency seems to be to use drop pipes which are intended to be frozen into the center of the cake of ice, because, by this method, it is possible to obtain more agitation than if the drop pipe has an opening only at the bottom. This agitation of the water, first mainly at the bottom of the pipe and then, later, through the holes along the length of the pipe, seems to have the effect of decreasing the amount of white core in the completely frozen cakes of ice.

The compressed air which is supplied for the agitation of the water in ice cans should preferably be taken from the air space in the freezing tank, that is, near the top of the tank, for the reason that the air in this space does not circulate much and is, therefore, cooled to a low temperature by the brine. This provision for obtaining cool air for water agitation in the cans is a worthwhile consideration for the best efficiency of an ice-making plant and is especially important in summer weather, when the air taken from the atmosphere for the same purpose would be at a very much higher temperature and would contain a great deal more moisture. The effect of injecting warm air and warm-water vapor into the ice cans is to increase the refrigerating effect required and to increase also the time required to freeze a cake of ice. The compressed air is usually delivered to the cans of ice by means of large distributing pipes with a great many outlets for the attachment of shorter lateral pipes. Each of the laterals has a number of outlets for the attachment of pieces of hose which run to the drop pipes in the individual cans. A typical arrangement of piping is shown in Fig. 218. Such an arrangement makes it possible to have a fairly uniform air pressure in all the cans of a freezing tank.

If the drop pipe is located in the center of the ice can, the temperature of the air in the pipe will be at 32° F. from the time that freezing begins until the water in the core is frozen. Under these conditions, with ordinary freezing temperatures in the drop pipe for practically the whole time that the ice can is in the freezing tank, no opportunity is given for the moisture in the air supplied for agitation to freeze in the drop pipe. In the high-pressure air-agitation system the drop pipe is mechanically attached to the side of the ice can either by being soldered in the corner, as in Fig. 218, or fastened in some other way to the side of the can, it is in metallic contact with the side of the can during the whole period of freezing of the ice cake and is, therefore, for

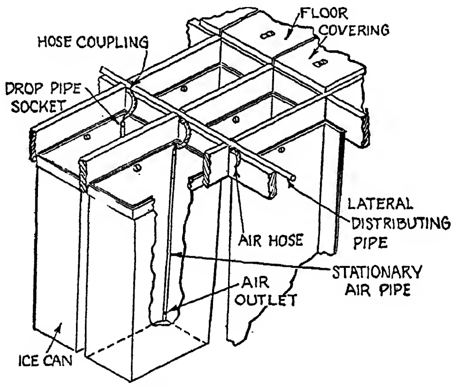


Fig. 218.—High-pressure air piping for ice cans.

all this time, at a temperature between 14 and 18° F. The air supplied through the drop pipe for water agitation under these conditions requires a higher pressure than when the drop pipe extends down through the vertical axis or middle of the can. This compressed air at the higher pressure requires the removal of some of the moisture in order to prevent the freezing of the moisture in the air in the drop pipe long before the ice cake is frozen. For this kind of air distribution for water agitation in the cans, it is customary to provide air in the lateral distributing pipes at somewhere between 10 and 20 pounds per square inch gage pressure, and, consequently, the compressor supplying this air will have to operate at a still higher pressure, usually from a few to 10 pounds more than the pressure in the laterals. All of these systems of

water agitation by means of compressed air require the removal of the water vapor, with the exception of the systems using the low-pressure system. The apparatus used for removing the moisture from air is called a *dehumidifier*.

Dehumidifier.—One of the simplest and most easily explained devices for removing the moisture from the air used for agitating the water in ice-making plants utilizes sprays of cold water and brine to chill it, the moisture in the air, when sufficiently chilled, being easily deposited and removed. This apparatus, called a *dehumidifier*, consists of two vertical cylindrical shells, the air, from which the moisture is to be removed, passing first through one and then through the other. In other words, the moisture

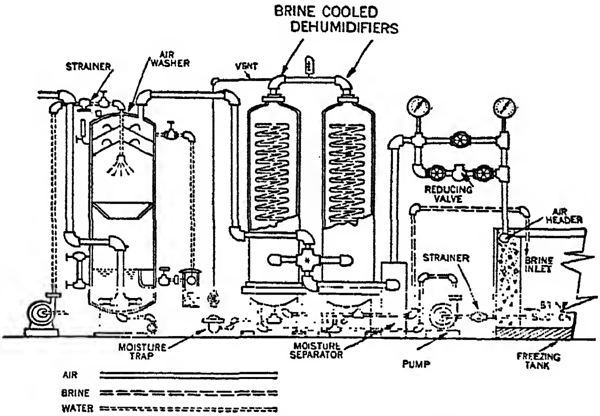


FIG. 219.—Brine-cooled dehumidifier.

to be chilled passes through these two shells in series. The shell through which the air first passes is usually about half full of water, which is cooled by means of a brine coil; and the second shell is half full of brine, which is cooled by the use of direct-expansion piping connected to the evaporator of the refrigerating system. The pipe carrying the air from the dehumidifier to the piping system supplying air to the ice cans has, usually, a cartridge type of air filter which is replaced every few hours.

The cartridge filter is necessary to collect the mineral matter remaining in the air after the moisture has been removed. This mineral matter must be taken out of the air in order to prevent its closing the very small orifices in the needle valves at the drop pipes supplying the air to the ice cans.

Another device for removing the moisture from the air required for water agitation consists of two vertical cylindrical shells arranged in series in the same way as explained for the preceding method with a coil of pipe in each shell (Fig. 219). Each of the two shells is cooled by the circulation of brine through the coils. The passage of the air through the first shell has merely the effect of reducing its temperature, and there is practically no frosting. In the passage of the air through the second shell, however, the moisture which is removed from the air collects as frost on the cooling surfaces of the coils and must be defrosted every 6 to 8 hours. This defrosting is accomplished by simply reversing a four-way valve. The advantage of this type of apparatus is that there is no dilution of the brine by the absorption of moisture, as in the first method. The accumulation of frost on the second coil assists in cooling the air by the amount of the latent heat of fusion of ice when defrosting the coil.

Power and Refrigeration Requirements for Air Agitation.—The amount of air required for the agitation of water in ice cans is quite large, so that air agitation in ice making involves a considerable operating expense. The first cost of the equipment for this service is also a large item. For the system of high-pressure air agitation in the ice cans, it is estimated that the usual power requirement is from 3 to 4 kilowatts[1] for every one-hundred 300-pound cans. In connection with air agitation, some refrigeration must be supplied to cool and remove the moisture from the air required for agitation. The low-pressure system requires usually only about 0.75 kilowatt per one-hundred 300-pound ice cans. In either system, about ½ kilowatt is required per one-hundred 300-pound cans for the operation of the core pump and the water and brine pumps. There seems to be a tendency, in the most recently constructed plants, to use a medium air pressure for water agitation in the cans, the gage pressure being about 10 pounds per square inch; and the drop pipe is then preferably in the vertical axis of the can, is made of brass, and extends nearly to the bottom of the can. In the low-pressure system, the drop pipe when centrally located, extends, usually, not nearly so near

[1] A further estimate might be stated here to the effect that the high-pressure system of air agitation requires about 6 kilowatt-hours per ton of ice, which is approximately one-eighth of the entire power requirement of an ice-making plant. It may be added that the services of one man are required to take care of the freezing-tank room per shift for every 60 tons of ice produced per 24 hours.

the bottom of the can and, in most cases, has its lower end about 9 inches from the bottom.

The medium-pressure system produces cakes of ice with very small cores, requires considerably less labor in the operations connected with the freezing tank, and does not require much more power than the old-fashioned low-pressure systems.

Removal of Core Water from Ice Cans.—There are not many kinds of natural water free from some kind of mineral matter which must be removed from the ice cans in order to make cakes of transparent ice. The tendency is for the particles of any kind of solid matter in water used for ice making to accumulate near the vertical axis of the cake of ice, in the part of the cake called the *core*. The water accumulating in the core has, in many cases, a taste somewhat like that of brine. This briny taste is due to the mineral matter. Some minerals have, also, the effect of discoloring the core. In order to rid the core of an ice cake of the briny taste and coloring matter, it is necessary to use some means for removing this objectionable water. It is necessary to remove the water in the core usually once and sometimes twice from a 300-pound cake of ice. This can be done very efficiently with a core-syphoning apparatus consisting of a small centrifugal pump, directly connected and driven by an electric motor, and a suitable tank, provided with an ejector device for removing the core water by suction.

There are many kinds of natural water, particularly in the eastern states, which can be treated and filtered, without removing the core water, so as to reduce and change the nature of the mineral deposits until the small amount remaining is unobjectionable.

Hoists for Ice-making Plants.—The hoists used in refrigerating plants where ice is made may be operated by hand, by compressed-air motors, or by electric motors. The electrically operated hoist is the one preferred in most plants, particularly if the ice cans are handled in large groups. A typical electric hoist is shown in Fig. 220, where a group of eight ice cans is being handled at a time. In the figure is shown an uncovered portion of the ice tank in which a row of cans has just been placed.

Dip Tanks.—The type of automatic can dump which is shown in Fig. 222 is now being replaced in many plants by a *dip tank* (Fig. 221). In order to obtain a large heat transfer to the cans in the dip tank, the water may be agitated with compressed air from a high- or medium-pressure system. Such agitation of the water

in the dip tank secures, also, a more even distribution of temperatures than would be possible without the agitation and promotes

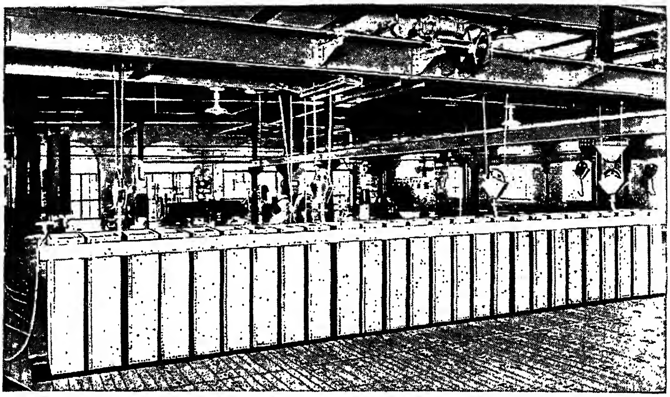


Fig. 220.—Electric hoist for ice cans.

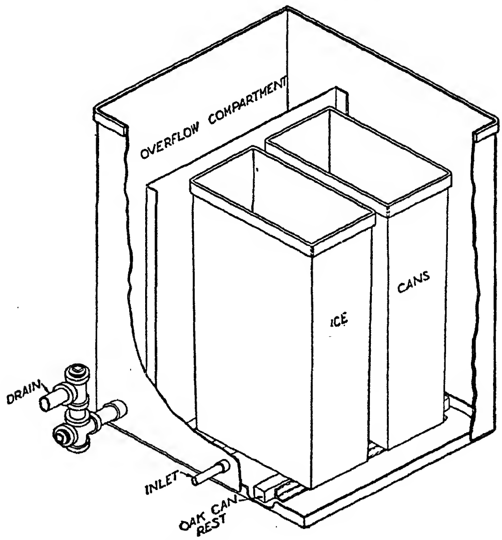


Fig. 221.—Dip or thawing tank.

the efficient operation of the dip tank by removing rapidly the cold film of water which tends to accumulate around the cans.[1]

[1] Tests have shown that the time required for thawing in the dip tank can be reduced about 40 per cent by the introduction of air agitation.

Ice-can Dumps.—Figure 222 shows an excellent design of an automatically operated ice-can dump, which can be used to very good advantage in reasonably large ice-making plants laid out to be operated with the group handling of ice cans. This

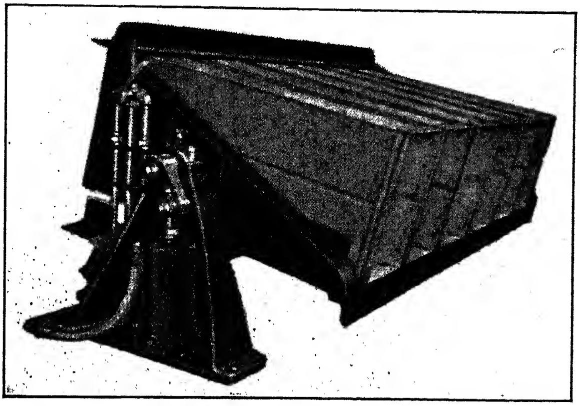


FIG. 222.—Automatic can dump.

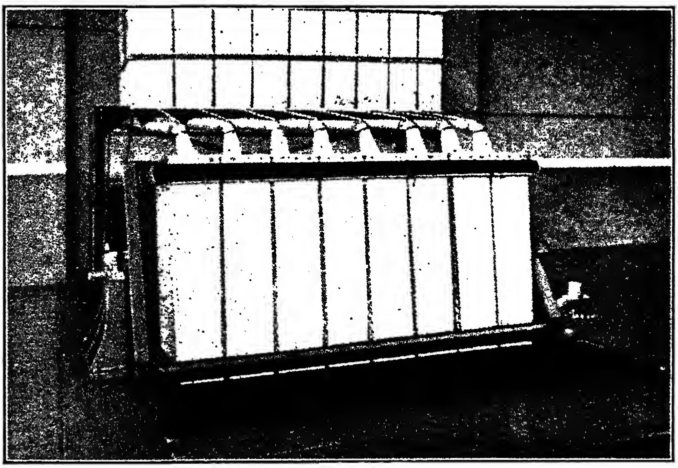


FIG. 223.—Ice cans in position for filling.

automatic dump is operated by a hydraulic cylinder *H* at one side of it. In the upright position of the dump, the ice cans are in the position to be filled with water, as illustrated in Fig. 223, while, in the position shown in Fig. 222, the ice cans are inclined

with the top sloping downward, so that the thawing water discharges over the sides of the cans and the cakes of ice when loosened will fall by gravity from them. Figure 224 shows a typical equipment for handling several ice cans at a time. The crane and hoist for group handling are shown overhead. A can-filling tank, shown to the right of the crane, discharges purified water into the individual cans, which are supported on an automatic can dump.

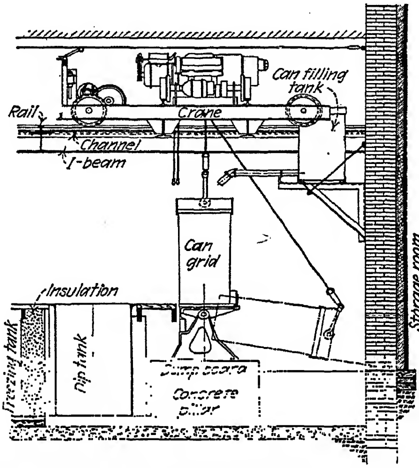


FIG. 224.—Crane and hoist for group handling of ice cans.

High-pressure Air Supply for Ice Cans.—A device for discharging high-pressure air into ice cans is represented in Fig. 218. The ice can shown in the figure has a heavy brass pipe extending down the side nearly from top to bottom. This pipe is provided to carry high-pressure air into the bottom of the ice can, the air entering the can about 1 inch above the bottom. There is a groove in the side of the can into which the pipe is set. The pipe is soldered at the lower end and is fastened to the side by slip ferrules, as shown. These slip ferrules permit expansion and contraction of the pipe. At the top of the can, it is fastened by soldering to a heavy brass socket which makes a tight joint with a suitable pipe fitting on the air line supplying the system.

A typical arrangement of piping for a high-pressure air system is shown in Fig. 218. This has the advantages over any of the low-pressure systems in that in the high pressure system it is not

necessary to remove the air pipes from the ice cans at all; thus a stopping of agitation is avoided and the consequent formation of a core in the ice cake, and, likewise, labor is saved. Thomas Shipley stated that the high-pressure air system produces much better results than a low-pressure system, "even though the low-pressure system is relatively satisfactory with proper care and with some kinds of water for ice making."

It is stated, further, that with a high-pressure air service and automatic can dumps having provision for handling with an

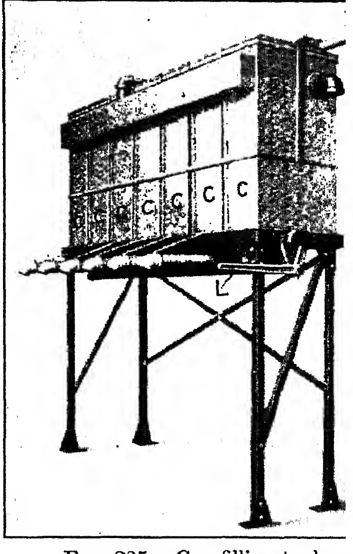


Fig. 225.—Can-filling tanks.

electric crane groups of eight or ten cans at a time, the time required to withdraw the cans from the ice tank, carry them with the crane to the automatic dump, remove the ice from them, and then refill them with purified water is only 7 minutes, the service of only one man being needed.

Can-filling Tanks.—For filling the cans in which the ice is made, a very simple device is shown in Fig. 225. This device consists of a row of cans C, each having the same water capacity as the cans in which the ice is to be made. The purified raw water flows into the filling tank through the pipe P, shown in detail in Fig. 226. This filling pipe is provided with a float

valve V operated by the float F, in one of the several chambers of the tank. After passing through the float valve, the water discharges into a trough T, which has holes near the bottom for discharging it into the compartments of the tank. A large outlet pipe is connected to the bottom of each of the compartments of the tank and has quick-opening and -closing valves all of which are operated by single lever L at the right-hand side of the tank shown in Fig. 225. It is desirable that the pipes for filling the cans in which the ice is made should have flexible connections to the valves to avoid injuring them in case the carrier of the crane handling the cans should interfere with them.

The holes through which the water discharges from the trough T (Fig. 226) into the compartments of the tank are provided

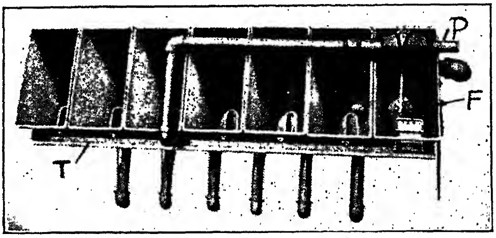


FIG. 226.—Automatic can-filling apparatus.

with nozzles, which make an even distribution of water to all the cans. These nozzles are so constructed that changes in the quantity of water delivered to the cans in which the ice is made can be easily made by raising or lowering them. Figure 223 shows a row of cans as they are being filled with water from one of these tanks.

Location of Direct-expansion Piping.—In some of the large plants which have been laid out for the manipulation of the ice cans by the "basket" method, it has been found necessary to place the direct-expansion piping on the bottom of the tank or to eliminate this piping altogether by the use of the shell-and-tube brine cooler.

A very recent arrangement of direct-expansion piping in an ice plant is shown in Fig. 212.

Operating Costs with "Basket" Arrangement of Ice Cans and Automatic Can Dumps.—The basket system of handling large numbers of ice cans at one time will probably save 30 per cent of the labor charge incurred when they are handled two at a time

with the can "dogs." The freezing tank can be made 10 per cent smaller when the basket system is used, and the cost of repairs is 50 per cent less than when the cans are handled with an old-fashioned two-can hoist. Of importance also, is the fact that there are fewer accidents when the basket system is used. It is stated that one man can lift one basket of ice cans (24 to 30 in a row), place them on the can dump, attend to the thawing, remove the cakes of ice from the cans, and refill them in less than 10 minutes.

Dual-pressure Ice-making System.—The operating economy of an ice plant may be greatly improved by the use of a multiple-

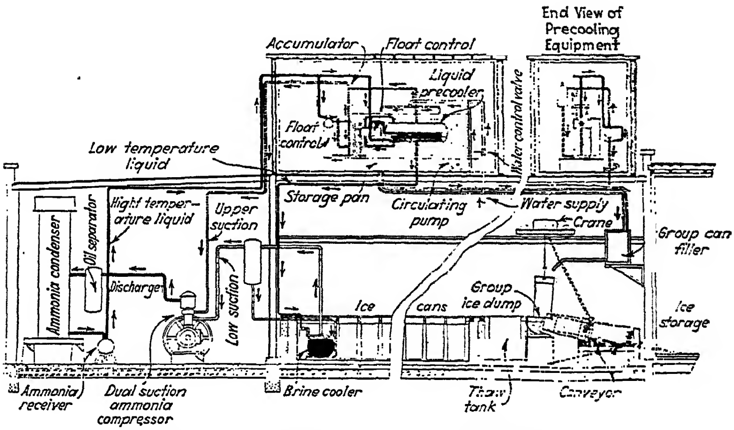


FIG. 227.—Dual-pressure ice-making system.

effect or *dual compressor* (p. 124). The arrangement of such a plant is shown in Fig. 227. The usual equipment is used except that in addition to the dual compressor a liquid ammonia pre-cooler and a water cooler are operated at the higher suction pressure. Each of these coolers is controlled by means of a float chamber. This equipment is usually placed in a small penthouse under the roof of the building. The water cooler can be of any type desired, such as a Baudelot cooler (p. 476) or other suitable equipment. The water after being cooled flows to the can filler while the cold ammonia liquid discharges into the brine cooler in the freezing tank. The method of handling the ammonia vapor from the water cooler and from the brine cooler is clearly shown in the diagram. A water-circulating pump continually

keeps the water to be cooled circulating over the Baudelot water cooler while a float valve controls the quantity of water supplied to the storage pan.

Power for Ice-making Plants.—In practically all the applications of refrigeration, the type of motive power for driving the compressors has changed from steam engines to electric motors and oil engines, the latter being usually of the Diesel type. When steam engines are used in ice-making plants, it is often convenient to use the condensation from the exhaust steam as the source of water to be frozen, as the water obtained in this way is suitable for use in plants requiring distilled water for ice making.

As the cost of labor and fuel has increased and it has become possible to distribute electric power more and more cheaply, nearly all modern plants near supplies of cheap electric current have come to have their compressors and auxiliary equipment operated by electric motors, because the compressors of an ice-making plant can usually be operated at a constant load. Synchronous electric motors[1] are peculiarly suitable for this service.

With the shift from steam-engine drive to electric motors and oil engines, obviously, the usual source of most of the distilled water for ice making was lost, so that recent ice-making plants have had to be designed for the use of raw water.

Details of Ice-making Plant.—As stated previously, there are several different methods of producing raw-water ice, but the general designing principles are the same. Engineers arrange the equipment according to the conditions. An Arctic Pownall raw-water ice plant, having a capacity of 40 tons of ice per 24 hours, is shown in Fig. 228. It should be noticed that the steel freezing tank is divided into eight compartments, each containing a definite number of cans. These cans are filled by opening one valve, which permits the water from the storage tank at a temperature of 38 to 40° F. to fill the cans. The storage

[1] The electric power that is transmitted and sold for power purposes is usually alternating rather than direct current. The two types of alternating-current motors commonly used are either induction or synchronous. A synchronous motor is generally preferred for driving compressors because of its high efficiency, and electric-power companies prefer to furnish electric current for this kind rather than for induction motors, the preference being due to the superior power factor of the former. Relatively few steam engines are now being installed in new refrigerating plants except in unusual circumstances.

tank is placed in the ice-storage room. Each compartment can be controlled separately, as each has two valves to control the flow of the brine. A vertical agitator is used to keep the brine in circulation. The more rapid the circulation the greater the transfer of heat.

The freezing tanks should be well insulated. It is common practice to insulate the bottom with 6 inches of cork board, which may consist of three layers of 2 inches each. The sides of the tank may be insulated with either cord board or granulated cork. In general, 12 to 18 inches of granulated cork is used, an

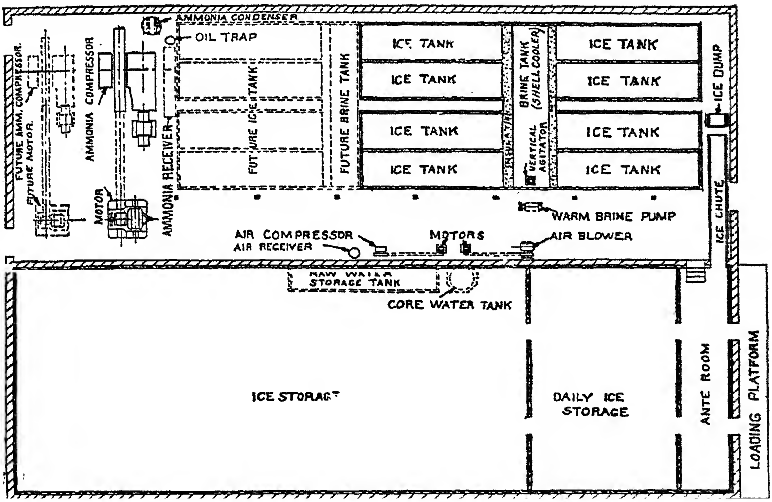


FIG. 228.—General plan of raw-water ice plant.

equivalent insulation of 6 to 9 inches of cork board. The bottom of the tank is insulated with 5 inches of cork board and 1 foot of cinders. Figure 229 shows typical construction of a freezing tank.

When a single ice tank is ready to harvest, the brine in this plant is warmed by cooling the storage water for the next filling. This process frees the ice from the cans. The ice is then drawn out and stored in the anteroom or, if for only daily storage, is placed in the "daily" ice-storage room. It often happens that, for certain reasons, the day's output is more than the demand. The surplus is frequently placed in the storage room, where it is kept at a temperature of about 24 to 28° F.

Shell-and-tube Brine Cooler.—The method of cooling the brine may be by direct-expansion coils placed in the tank, as shown in Fig. 210. Another method is to cool the brine with a shell-and-tube brine cooler shown in Fig. 213. In Fig. 228, this is located in the center of the tank, dividing the freezing tank into two sections. The cooler itself resembles a return fire-tube boiler. It is cylindrical in shape, having a shell through which a large number of tubes pass. The shell brine cooler is nearly filled with liquid ammonia. The liquid ammonia enters at the bottom of the cooler, and the ammonia vapor is removed from the top. The brine is circulated through the tubes and about the shell. The brine velocity in the tubes is about 2 feet per second. At higher velocities the agitator-horsepower goes up rapidly. In

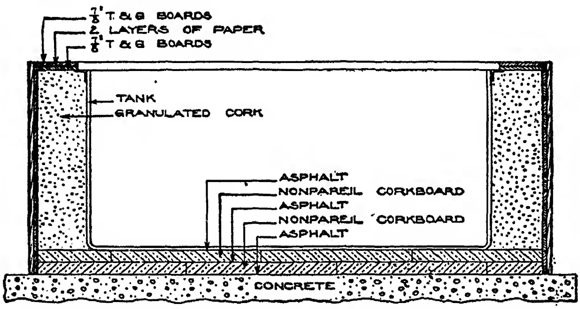


FIG. 229.—Typical construction of ice-freezing tank.

this system, the bulkheads are rigidly fastened to the bottom of the ice tanks, thus forcing the brine to travel in a positive direction and producing an even flow throughout the tank. This arrangement gives a uniform freezing rate.

It should be remembered that the length of time allowed for freezing is governed by the temperature of the brine and, also, by the rate of brine circulation. The brine is generally held at about 12 to 18° F. depending upon the season. If the rate of freezing is to be increased, the brine temperature must be lowered. In order to lower the temperature, a lower suction pressure is necessary. If the ice is made at a temperature too low, it will crack when the cans are placed in the dipping tanks. This cracking produces unmarketable ice.

With brine at a temperature of 14° F., it will take about 51 hours to freeze a 300-pound cake. If the temperature is

decreased to 12° F., a period of about 47 hours is required. By increasing the rate of brine circulation and by keeping the brine at a given temperature, the rate of freezing can be greatly increased.

Refrigeration per Ton of Ice for Varying Water Temperatures.—The heat removed to freeze ice is made up of the heat required to cool the water to the freezing point, the latent heat of fusion, the heat removed to bring the ice down to the temperature of the brine, and the heat loss from the freezing tank, cans, and covers.

If the initial temperature of the water is 72° F., the temperature of the brine is 14° F., and the losses are about 20 per cent of the actual refrigeration, the amount of refrigeration required to produce one pound of ice is calculated as follows:

Heat removed to cool water	: $1(72 - 32)$	$= 40$	B.t.u.
Latent heat of fusion	: 1×144	$= 144$	B.t.u.
Heat removed to cool ice	: $0.5(32 - 14)$	$= 9$	B.t.u.
Total heat without losses		$= 193$	B.t.u.
Losses	: 193×0.20	$= 38.6$	B.t.u.
Total heat required per pound of ice		231.6	B.t.u.

In order to obtain one ton of ice it will be necessary to produce

$$\frac{231.6 \times 2{,}000}{288{,}000} = 1.61 \text{ tons of refrigeration.}$$

If the temperature of the brine is 14° F., the number of tons of refrigeration may be found from the following table:

TABLE XIII.—TONS OF REFRIGERATION PER TON OF ICE

Initial temperature of water ° F.	Tons of ref. per ton of ice	Initial temp. of water ° F.	Tons of ref. per ton of ice
40	1.34	62	1.53
42	1.36	64	1.55
44	1.38	66	1.56
46	1.39	68	1.58
48	1.40	70	1.59
50	1.42	72	1.61
52	1.44	74	1.62
54	1.46	76	1.64
56	1.47	78	1.66
58	1.48		
60	1.50	80	1.68

Time of Freezing of Ice.—The time of freezing ice varies with its thickness, the temperature of the brine, the shape of the can or mould in which the ice is formed, the temperature of the water, and circulation of the brine. The most important factors for a given mould are the mean difference of temperature and the thickness of the ice formed. An empirical formula has been used to determine the freezing time for *can* ice as follows:

$$\text{Freezing time in hours} = t_f = \frac{7 \times t^2}{32 - t_b}$$

where t = thickness of ice in inches
t_b = temperature of brine, degrees Fahrenheit.

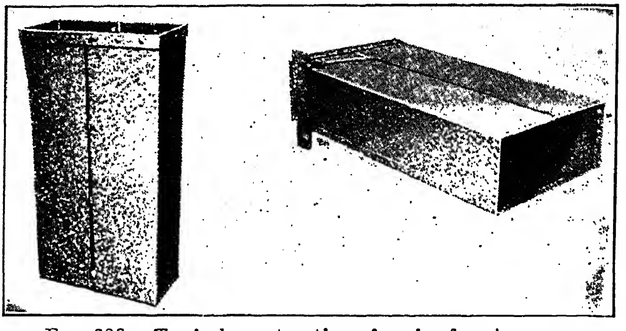


Fig. 230.—Typical construction of an ice-freezing can.

In the case of the *plate* system the above formula is slightly modified and becomes $t_f = \dfrac{21 \times t^2}{32 - t_s}$ where t_s is the temperature of the freezing surface.

Example: Find the number of hours required to freeze a standard 300-pound cake of ice if the brine temperature is 14° F. The dimensions at the top of a 300-pound ice can are $11\frac{1}{2}$ inches × $22\frac{1}{2}$ inches × 46 inches. The average thickness is 11 inches and substituting in formula above, the freezing time in hours is

$$t_f = \frac{}{32 - 14} = 47 \text{ hours.}$$

Construction and Size of Ice Cans.—Ice cans are made with a taper so that the ice blocks may be easily removed without

an excessive loss due to thawing. The cans are made of galvanized iron and have a rectangular cross-section as shown in Fig. 230. The joints are either welded or riveted, and the top is provided with a band to stiffen it.

The standard dimensions for ice cans used in the United States are given in the following table:

TABLE XIV.—DIMENSIONS OF ICE CANS

Weight of cake of ice, pounds	Inside dimensions, inches			Length over-all, inches	Thickness of material—U. S. Standard gage
	Top	Bottom	Length		
25	4 × 9	$3\frac{1}{2} \times 8\frac{1}{2}$	23	24	No. 18
50	5 × 12	$4\frac{1}{2} \times 11\frac{1}{2}$	31	32	No. 16
50	6 × 10	$5\frac{1}{2} \times 9\frac{1}{2}$	31	32	No. 16
50	8 × 8	$7\frac{1}{2} \times 7\frac{1}{2}$	31	32	No. 16
60 (25 kilograms)	5 × 14	$4\frac{1}{2} \times 13\frac{1}{2}$	31	32	No. 16
100	8 × 16	$7\frac{1}{4} \times 15\frac{1}{4}$	31	32	No. 16
200	$11\frac{1}{2} \times 22\frac{1}{2}$	$10\frac{1}{2} \times 21\frac{1}{2}$	31	32	No. 16
200	$14\frac{1}{2} \times 14\frac{1}{2}$	$13\frac{1}{2} \times 13\frac{1}{2}$	35	36	No. 16
300	$11\frac{1}{2} \times 22\frac{1}{2}$	$10\frac{1}{2} \times 21\frac{1}{2}$	44	45	No. 16
300	11 × 22	10 × 21	48	49	No. 16
400	$11\frac{1}{2} \times 22\frac{1}{2}$	$10\frac{1}{2} \times 21\frac{1}{2}$	57	58	No. 14
400	11 × 22	10 × 21	61	62	No. 14

Number of Cans Required in Freezing Tank.—In order to freeze one 300-pound can of ice with a brine temperature of 14.37° F. a freezing period of 48 hours will be required. From this it is seen that two ice cans must be in the freezing tank for each can that is harvested.

The number of cans per ton of ice varies indirectly with the brine temperature, and for standard 300-pound cans there will be needed 13.3 cans per ton of ice. This relationship can be expressed by the following formula in which N is the number of cans per ton of ice.

$$N = \frac{2{,}000 \times t_f}{w \times 24}$$

where t_f is the freezing time in hours and w is the weight of an ice cake in pounds.

This formula may be simplified by substituting for w the various weights of the ice cakes formed in standard cans.

$$200\text{-pound can, } N = \frac{t_f}{2.4}$$

$$300\text{-pound can, } N = \frac{t_f}{3.6}$$

$$400\text{-pound can, } N = \frac{t_f}{4.8}$$

In order to operate the ice plant economically, it is important to design the plant for the proper number of cans per ton of ice. If the number of cans per ton of ice is small, a low brine temperature will be needed to freeze the ice in the required time. This low brine temperature will necessitate a low suction pressure. On the other hand, if a large number of cans per ton of ice is used, which means a large number of cans for a given tonnage output, the initial cost will be too great. This will, of course, raise the brine temperature and suction pressure which will produce better operating conditions.

As the brine temperatures commonly used have a range of 10 to 20° F., the number of standard 300-pound cans per ton of ice will vary from about 10 to 20 cans. A general rule is fourteen 300-pound cans per ton of ice for distilled water ice plants, while sixteen 300-pound cans per ton of ice is used for electrically driven raw-water ice plants. These values closely correspond to brine temperatures of 14 to 16° F.

Direct-expansion Piping for Ice Freezing Tanks.—The calculation of the surface needed in an ice tank depends on the type of cooling; that is, whether the cooling is by direct-expansion system, flooded direct-expansion system or shell-and-tube brine cooler. The shell-and-tube brine cooler may be considered as a flooded system.

The amount of surface needed depends upon the refrigeration required to produce one ton of ice, the heat coefficient of the cooling surface, and the mean difference in temperature of the refrigerant and the brine.

As previously pointed out about 20 per cent is allowed for losses (see p. 349), but it often happens that there is an additional amount of refrigeration needed, as the ice storage house and anteroom are often cooled by brine taken directly from the freezing tank.

Heat-transfer coefficient C for a dry direct-expansion system is about 15 B.t.u. per square foot per hour per degree Fahrenheit difference in temperature, for a flooded system, 20 to 30 B.t.u. per square foot per hour per degree Fahrenheit difference, and for a shell-and-tube brine cooler the heat-transfer coefficient is about 80 to 100 B.t.u. per square foot per hour per degree Fahrenheit difference in temperature. The trunk system is designed to give a heat-transfer-coefficient range of 90 to 120 B.t.u. per square foot per hour per degree Fahrenheit.

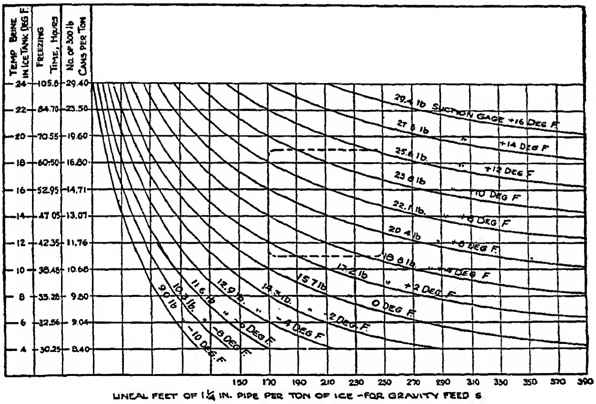


FIG. 231.—Curves based on heat transmission coefficient of 25 B.t.u. per sq. ft. of surface per °F. per hour and 220 B.t.u. per pound of ice-cooling effect in freezing tank.

The square feet of surface required per ton of ice may be calculated from the following equation:

$$S = \frac{H}{24 \times C \times t_d}$$

where H is the heat removed per ton of ice in B.t.u., C is the heat transfer coefficient and t_d is the mean temperature difference in degrees Fahrenheit between the brine and the refrigerant.

If the temperature of the water is about 70° F. and 1¼-inch direct expansion pipe is to be used, the above formula may be written in the following way:

$$\text{Linear feet of 1¼-inch pipe} = \frac{288{,}000 \times 1.6 \times 2.3^*}{24 \times C \times t_d} = \frac{44{,}160}{C \times t_d}$$

* 2.3 linear feet of 1¼-inch pipe are equivalent to 1 square foot of external surface.

In order to simplify the work of finding the linear feet of $1\frac{1}{4}$-inch pipe per ton of ice, Fig. 231 may be used. The curves are based on an ice-cooling effect of a 220 B.t.u. per pound and are for the flooded direct-expansion gravity-feed system. A heat-transfer coefficient of 25 B.t.u. per square foot per hour per degree Fahrenheit difference in temperature was used in calculating these curves.

The mean difference in temperature between the brine and the refrigerant is determined by economical considerations. This difference will vary between 6 and 12° F. It should be noted that if a small difference in temperature is used, a larger surface will be needed than with a larger temperature difference. The initial cost of the piping will, therefore, depend on the temperature difference. With a small temperature difference the cost of the power will be less as the suction pressure will be higher.

Surface Required for Shell-and-tube Brine Cooler.—If a shell-and-tube brine cooler is to be installed instead of direct-expansion piping, the area of the brine cooling surface in square feet per ton of ice can be calculated from the following equation:

$$S = \frac{H}{24 \times C \times} \qquad \frac{288{,}000 \times 1.6}{24 \times C \times t_d} - \frac{19{,}200}{Ct_d}$$

Brine coolers have been surfaced to give a 3 to 4° F. difference in temperature between the brine and the ammonia.

Ice Storage.—The production of ice is largely a seasonal business, the demand varying considerably with the weather conditions and is particularly heavy at the week end. In order that the plant may run on a regular output, some storage space is necessary, not only to take care of the surplus when the demand is small but also to take care of the heavy demand due to hot periods, week ends, and holidays. In general, the ordinary plant should have a storage room which is capable of storing from three to ten times the plant output per day. The floor area for the "day"-storage room may be determined by allowing about 50 cubic feet of space per ton of ice when stacked, but since it is customary to place the ice cakes on end in the day-storage room an area of 14 square feet per ton of ice for 300-pound cakes and 11 square feet per ton of ice for 400-pound cakes should be allowed. The ice cakes are stacked four to seven tiers high in day storage.

The walls and roof of buildings for ice storage should be insulated with cork board laid in asphaltum. Floors may be insulated

to a depth of 36 to 42 inches of well-tamped soft coal cinders, over which an oak floor is laid with ¼-inch spacing. The oak floor is nailed to 2- × 3-inch wood screeds imbedded in the cinders. This type of floor insulation is a good preserver of ice and is exceptionally dry. In some storage rooms, 12 inches of mill shavings are used for insulating the outside walls, with 24 inches of shavings over the ceiling or under the roof. The temperature of ice storage rooms should be from 24 to 28° F., depending on the depth of the storage.

There is no difficulty in stacking ice to a considerable height in the main storage house as there is no in or out traffic, the house slowly filling in the winter and slowly emptying in the summer. For raising the ice cakes, lifts or hoists can be used, while the lowering of the ice cakes may be accomplished simply by the use of a spiral conveyor. The cakes of ice should be laid with the large and small ends alternating in each layer, so that the surface remains nearly level. Stacking should be arranged so as to tie the layers together to prevent sliding.

In most ice storage rooms direct-expansion piping is used. In some plants, however, brine is taken from the freezing tank and pumped through the coils in the ice storage.

Flakice.—Ice that is made from a good grade of water which is formed so as to have *curved* surfaces and has a thickness of about ⅛ to $\frac{3}{16}$ inch is known as "flakice." It has some advantages as it can be made cheaper than can ice, and has some new applications. The surfaces being curved cannot freeze together, as they come in contact only at points or lines, thus making voids between the flakes; the larger the flakes the greater the volume of the voids. Flakice about ⅛ inch in thickness, if broken by a fall of about 10 feet, has a void volume of about 20 to 30 per cent. The appearance of flakice depends on the freezing temperature; if the refrigerant temperature is low, the ice flakes are white; while, if the freezing temperature is higher, it is transparent. The amount of air used for agitation (p. 334) in this type of ice plant is about one-half to one-quarter of that required to produce can ice.

Present applications are in restaurants, ice-cream plants, meat and fish markets, confectionery stores, dairies, chemical plants, hospitals, and air-conditioning equipment. In a test made on an ice-cream freezer, 100 pounds of flakice was required where 192 pounds of crushed ice was normally used. In air conditioning

or in chemical processes, the rate of melting is practically constant until approximately 90 per cent of the ice flakes are melted.

The machine for making flakice (Fig. 232) consists of a distorted cylinder the shape of which is caused by the weight of the brine inside the cylinder, the water outside of it, the deflecting roller at the top, and the guides. This cylinder is made of seven metal panels separated by rubber strips or bands and end aprons. The function of the metal is to provide a rapid heat-transfer surface on which the ice can form. The rubber strips bind the metal panels into a complete cylinder and serve as an insulation between the various panels thus dividing the ice sheet into sections as wide as each panel.

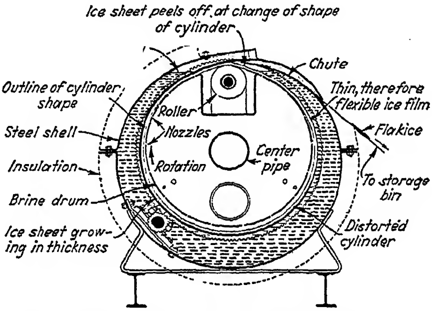


Fig. 232.—Machine for making "flakice."

Brine from a brine cooler is allowed to flow into the cylinder at one end and out the other. For this purpose the shaft of the cylinder is made hollow. The water level is maintained by means of a float valve located at a point just below the top of the horizontal cylinder. As the cylinder rotates slowly, the water quickly freezes to the surface in the form of a thin film which gradually increases in thickness. During the early part of the freezing, the ice film is quite flexible, but as it becomes thicker it peels off under the change in the shape of the surface of the cylinder. A chute is provided for catching the ice sheets and allows them to fall, thus breaking the sheets into flakes.

When water at a temperature of 40° F. is supplied to one of these machines a ton of ice can be produced by 1.2 tons of refrigeration.

Pakice Freezer.—Another type of ice-making equipment built for the purpose of producing either water or brine ice is the

"Pakice" machine. This freezer takes the place of the conventional ice-making tank and eliminates some of the necessary equipment used in a can-ice plant, such as the brine agitator, crane, and ice crusher. A brine ice made from a 23 per cent sodium-chloride solution can be made by this means which has a melting point of $-4°$ F. and is suitable for car icing and packing ice cream. Higher temperatures may be had by using weaker brine solutions. For example, a 15 per cent sodium-chloride brine has a temperature of 3° F. for a few hours which rises to 15° F. when half melted.

In this system no attempt is made to produce clear transparent ice. A mixture of 75 per cent ice and 25 per cent water may be packed in the ordinary ice can and frozen. The time required

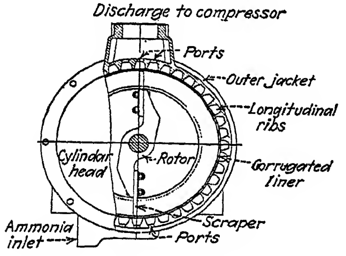


FIG. 233.—Cross-section of machine for making "pakice."

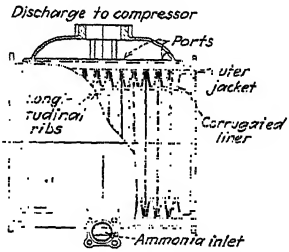


FIG. 234.—Longitudinal section of machine for making "pakice."

for freezing is about 12 hours with brine having a temperature of 17° F. Tests show that the product, as it comes from the freezer, when substituted for crushed ice is cheaper, easier to handle, and produces lower temperatures. The ice, as it comes from the freezer, is not in the form of flakes but in the form of small crystals which are suspended in water. A stream of water carries the crystals from the freezer and deposits them in a storage bin. The water used for transporting the ice crystals is then drained from the bottom of the bin and re-circulated through the freezer. Make-up water is added to the system as fast as the ice is removed. Two cross-sectional views of this freezer are shown in Figs. 233 and 234. This freezer consists of an outer casing within which is placed a corrugated liner. Liquid ammonia circulates in the space between the two surfaces. The inside of the liner is filled with circulating water which freezes rapidly on the liner surface and is removed by means of rotating steel

scrapers. The rapidity of freezing is such that 15 gallons of water is frozen in less than 4 minutes. The ice that is cut loose from the liner is driven toward the center where it mixes with the re-circulating water and is carried out of the freezer. The rate of heat transfer is about 300 B.t.u. per square foot per hour per degree Fahrenheit. A liquid head of 4 feet is maintained on the ammonia thus keeping the casing flooded and causing the ammonia to circulate against the outer surface of the liner.

Freezing-tank Design for Can Ice

Size and shape of tank:

a. Controlled by the size and shape of the building.
b. Consideration in laying out tank should be given to the harvesting equipment, so as to reduce labor costs to a minimum.
c. Short tanks are commonly used today.

Number of cans to be installed daily per ton of ice:

a. Depends upon the efficiency of tanks.
b. Sometimes chosen so as to give high brine temperature with a given back pressure; sometimes low temperatures are used but must not be so low as to produce cracking.
c. An economical figure is 7.5 cans of 100 pounds capacity per ton of ice, back pressure of 25 pounds per square inch gage, 14° F. brine, and a temperature head of 2.7° F.; that is, the temperature of the brine would be 2.7° F. above that of the ammonia in the brine-cooling coils. This gives a heat-transfer value for external surface of about 46 B.t.u. per square foot per hour per degree Fahrenheit.

Advantages of flooded system:

a. Expansion valves eliminated.
b. Constant level of ammonia in accumulator.
c. Liquid always available for coils and not controlled by a hand-operated expansion valve.
d. Fluctuating condenser pressures would not affect the flow of liquid through the coils.
e. The internal coil surface should be wetted by liquid ammonia to the fullest extent, provided that the coils are of the proper length.
f. Condensers should be drained automatically, and surplus liquid held in the accumulator and evaporating coils.

Coils:

a. Split coils should not be installed, as they do not produce an even amount of work, due to the liquid heat's being greater on the bottom coil.
b. The coils can be designed single, double, triple, or quadruple, depending upon the length of tank.

c. The length of each individual coil should not exceed 220 feet when 1¼-inch pipe is used.

Approximately 250 feet of 1¼-inch pipe would be required per ton of ice per day.

Economy:

a. The economy of the ice tanks can be improved by increasing the rate of brine circulation; this will increase the transfer of heat from the can surface to the brine cooling coil surface.

Coefficient of heat transfer for coils (flooded system):

a. Old figure, 15 B.t.u. per square foot per hour per degree Fahrenheit.
b. Government, 20 B.t.u. per square foot per hour per degree Fahrenheit.
c. Late plants, 24 to 28 B.t.u. per square foot per hour per degree Fahrenheit and in certain cases may be as high as 46 B.t.u. per square foot per hour per degree Fahrenheit difference.

Coil Spacing:

a. Present-day design, 14½ inches on centers.
b. Wall coils should be arranged so as to be 3 to 5 inches from the tank wall.

Can Arrangement:

a. Cans should be arranged so as to prevent short-circuiting of the brine.
b. Space below cans should be reduced to a minimum; in fact, the floor can be made solid beneath them.

Freezing-tank Design:

a. As a rule, tanks for 400-pound cans are 63 inches deep, but if high agitation is to be used, it will be necessary to make the wall on the "high-brine" side 68 to 70 inches high for some distance away from the propeller.
b. For fair agitation, the tank floor should slope 1 inch for each five cans lengthwise with the tank.
c. Capacity of plant can be increased by having the ice well submerged. For example, a tank designed to freeze ice in 48 hours, with a brine level even with the ice in the cans when the block of ice is completely frozen, would, however, freeze the block in 43 hours provided that the ice were submerged 5 inches.

INSULATION AND COLD-STORAGE CONSTRUCTION

Insulation.—Refrigeration consists in removing heat from substances and in keeping outside heat away from them. This work is made difficult by the fact that heat flows from places of higher temperature to places of lower temperature and that goods which have been cooled by refrigeration will again absorb heat unless care is taken that no heat shall come into contact with them.

No matter how well a refrigerated compartment is made, some heat is constantly passing into it from the outside. If a way is not found to remove this heat as fast as it comes in, the temperature of the compartment will gradually rise. It is necessary, then, always to carry on the process of refrigeration at such a rate as to offset the inflow of heat. If the temperature of the compartment rises $\frac{1}{2}°$ an hour by inflow, it is necessary to remove $\frac{1}{2}°$ an hour by refrigeration. Clearly, all the work of removing heat is expensive, and it is important that the inflow should be reduced as far as possible, to decrease the cost of refrigeration needed to offset it. For that reason, a good deal of attention is being given to the construction of chambers which will effectively keep out heat.

All that portion of the walls, floor, and ceiling of a refrigerated space, that tends to keep heat from entering, is called *insulation.* Of course, no insulation exists that can keep out all the heat, but certain kinds of insulating material can keep out more than others. On the quality of insulation and on the tightness of walls, windows, and doors depends the amount of heat that leaks into a compartment during a given time.

Since the purpose of insulation is to cut down the work of refrigeration, the engineer must consider the relative cost of insulation and refrigeration when he chooses a certain quality of insulating material. It would be poor business to spend more on high-grade insulation to keep the heat out than it would cost to remove that heat by refrigeration. As a rule, however, a

poor grade of insulation should never be used, for poor insulation results in uneven temperature in the compartment. Where the temperature is not uniform, goods in the center of the chamber are likely to freeze before those near the walls are cold enough.

Beside keeping out heat effectively, there are five other requirements that good insulating materials must meet:

1. It should be light so that it will not pack and settle, leaving the upper spaces of the wall unprotected.
2. It should not absorb moisture, for damp insulation permits the passage of heat more rapidly than the same material would when dry, and dampness causes decay and fermentation in the insulation itself.
3. It should be proof against disintegration and spontaneous combustion.
4. It should be odorless, so that it will not taint perishable goods, such as butter, stored in the compartment.
5. It should be proof against the tunneling and nesting of rats and other vermin.

Since few insulating materials are waterproof, it is generally necessary to enclose them in materials which are waterproof to prevent their absorbing moisture.

Materials.—We come now to a study of those substances so far classed as *insulating materials.* Among those used in cold-storage compartments and houses, the most common are sawdust, shavings, charcoal, mineral wool, still air, straw or chaff, paper, hair felt, quilting, wood, cork, celotex "dry zero," and aluminum foil.

Still Air.—Air is probably the most common insulator of all, because it conducts heat very slowly. It has been in use for a long time in guarding cold-storage compartments against inflow of heat from the outside. The most simple arrangement for insulating with air is a double-chamber wall with an air space between the inner and the outer wall. A room made in this way is surrounded by air spaces on all sides, but the spaces are so large that the air moves easily. Air which has been warmed by contact with the outer wall rises until stopped by the top of the air chamber. Here it is forced, by the rising air under it, over against the cold inner wall; it cools and travels down the inner wall to the bottom. Then the process begins all over again. But when it is said that the air is "cooled by contact with the

inner wall," the meaning is that it gives up some of its heat, which then passes through the inner wall and raises the temperature of the compartment. Thus, *moving air*, by absorbing heat from the outer wall and giving it to the inner wall, serves as a heat carrier instead of an insulator.

To keep air from circulating in the manner just described, crosspieces are sometimes nailed between the inner and the outer wall, to form smaller air cells, between 1 and 2 feet square. And, sometimes, instead of only two walls (an inner and an outer), there are three or four walls with air spaces between. Even with these precautions, however, the air circulates and fails to provide effective insulation.

It is possible to keep air motionless by packing the air space with loose materials. Among such materials are sawdust, straw, shavings, and nearly all of the other things that have been mentioned as insulators. Some of them are in themselves slow to conduct heat, but they are also valuable because they divide the air spaces into tiny cells in which there is practically no circulation of air. The filling material is packed firmly so that the cells shall be as small as possible; but it is not compressed, for if the particles of filling are in too close contact with each other, the heat may be conducted from one to the other and so to the inner wall.

The thermal conductivity of an ideal air space (p. 523) is about 0.175 B.t.u. per square foot per hour per degree Fahrenheit, while for an air cell ½ inch wide the conductivity is 0.458 B.t.u. per square foot per hour per degree Fahrenheit. In the case of a 1-inch air cell the conductivity is 0.5.

Insulating by Means of a Vacuum.—As already explained, air is a good heat insulator, but a still better insulator is made when, by the withdrawal of air from a space, a fairly good vacuum is established. A commonly used form of vacuum insulator is a so-called "vacuum bottle." The first of these was made about 1885 in Germany for the scientist Dewar; and, because of this introduction, such bottles are sometimes called "Dewar flasks."

Air *conducts* heat chiefly by the movements of molecules which at normal room temperatures move at the approximate speed of 1,500 feet per second, but in such zigzag ways that they are continually colliding with each other. It can be stated for a wide range of pressures that the heat conductivity of air is

practically independent of the pressure.[1] When, on the other hand, the pressure of air is very greatly reduced, quite different results are obtained. In fact, at very low pressures (high vacuums), the heat conductivity of air is, for practical purposes, directly proportional to the pressure.

At "ordinary" room temperatures the molecules of air have no particular use in the transmission of heat by *radiation*. When, however, the temperature is lowered, the frequency of the heat waves decreases and, therefore, the wave length increases. Because of this fact, heat radiation will take place across an almost perfect vacuum, but this sort of radiation can be reduced a great deal by the use of *silvered* reflecting surfaces which are so commonly used in thermos bottles. When this silvered reflecting surface becomes tarnished, the transmission through the vacuum of radiated heat is increased.

The conduction of heat through highly rarefied (low-pressure) gases across a double-wall container of the types ordinarily used depends on the area of the exposed surface, the difference in temperature between the two walls, and the degree of rarefaction of the gas. In a *Report* of the U. S. Bureau of Standards, the conductivity of the vacuum space with two silvered walls is given as 0.004 B.t.u. per square foot per hour per degree Fahrenheit.

Although there are not many practical uses of a vacuum for heat insulation, there are important examples of its use in the ordinary thermos bottle, sometimes made in large sizes, and in two-wall containers with a high vacuum between them.

Manufacture of Cork Board.—As cork board was first made, it was produced by cementing together the cork waste left over from the manufacture of stoppers for bottles and similar articles. As the uses of cork board increased, there was an increasingly large demand for this waste cork, and at the present time very little of this waste is now available for making cork board. For cork insulation by the method of filling with the granulated kind, the material used, whether cork waste or virgin cork, is likely to be of inferior grade, as the better quality of granulated

[1] When the atmospheric pressure is reduced, for example, to one-fifth of the normal pressure, the number of molecules is reduced in the same proportion and the "free path" for the molecules becomes five times as long. Since, therefore, the molecules have been decreased in number to one-fifth, but, at the same time the path of each one has been made five times as long, the rate of heat conduction is the same as before.

cork is required in large quantities for the manufacture of linoleum.

Re-granulated cork is different in composition from the granulated cork and is obtainable in two sizes or grades. It is made from the trimmings which remain when cork board is sawed into the standard sizes of sheets or blocks. Granulated cork is graded and sized to a relatively large number of standards; on the other hand, re-granulated cork is graded only as "fine" or "coarse." Fine re-granulated cork varies in size from particles of dust to granules sometimes as large as 1/4 inch on a side. Coarse re-granulated cork varies in size from 1/4 to 3/4 inch on a side.

Of all such insulating materials, granulated cork is one of the best. Besides being a good non-conductor of heat, it is light, slow to absorb moisture, durable, odorless, and not subject to spontaneous combustion. In fact, it has all but one of the qualities that a good insulating material should have: It is attractive to vermin.

Cork is produced from the bark of the cork tree, belonging to the oak family and grows in Portugal, Spain and also in other Mediterranean countries. The cork tree lives to a ripe old age and is first stripped of its bark at the age of 10 to 15 years. This first stripping is of little value being used only for ornaments. Seven years later the tree is again stripped and these curved strippings are flattened by weights under water and also charred or steamed in large copper boilers to close the pores.

The granulated cork is made from shavings and chips ground up in mills. Care is taken to remove the fibre from the bark which contains grit. The ground cork is then sifted and screened; the coarse and medium sizes are generally used for refrigerating purposes.

Cork is exceedingly light and it packs naturally to about 5 pounds to the cubic foot. But in order to prevent the smaller pieces from settling, granulated cork is generally packed to a density of about 6 pounds to the cubic foot.

Cork is porous; that is, it contains many tiny air spaces. If it is granulated, then, and packed between the walls of a refrigerating compartment, there will be particles of air *between* the grains and, still more, little air cells *in* the grains themselves. Thus, the insulating properties of air are added to the insulating properties of the cork itself.

Loose granulated cork can be packed lightly into the space between the inner and outer wall, but this arrangement leaves the grains unprotected against moisture and vermin. To overcome this objection, manufacturers compress granular cork in iron molds and apply heat at about 700° F. In this way, slabs are made, measuring 36 inches in length by 12 inches in width. The slabs have all the good qualities of loose cork but are actually more efficient as heat insulators.

In the formation of these slabs or cork boards as they are often called, the pressure and heat cause the natural resin to act as a binder which holds the particles of cork together.

The thermal conductivity for granulated cork is about 0.31 to 0.33 B.t.u. per square foot per hour per inch of thickness and per degree Fahrenheit. Generally, though, about twice the thickness of granulated cork is used to that of cork board. The thermal conductivity for cork board is about 0.25 to 0.31 B.t.u. per square foot per hour per inch of thickness and per degree Fahrenheit. These values change somewhat with the density which varies from about 6.9 to 10 pounds per cubic foot.

As a guard against moisture and rats, granulated cork is sometimes bound together with pitch or asphalt, but this mixture conducts heat more readily than either the loose grains or the

Where the refrigeration compartment is a brick-walled structure, cork slabs are simply laid against the wall after the bricks have been coated with hot pitch. They adhere without the use of further fastening. The inside of the slabs is usually finished with portland cement. In applying cork slabs to a frame wall, waterproof paper is laid on first; then furring strips are nailed on, and the slabs are placed between the furring. For this type of wall, the inside is finished with a layer of waterproof paper next to the cork, and a sheathing of matched and dovetailed boards is placed over that. Some kinds of slab are made up with a nailing strip imbedded in the center so that the slabs may be nailed to the wall or to the sheathing which covers them.

Hair Felt.—Another high-grade insulating material is hair felt. It is made of cattle hair which has been subjected to a careful process of washing and cleaning and compressed into a matlike substance by a felting machine. It is put on the market in sheets or strips varying in width from 2 to 6 feet and in thick-

ness from ¼ inch to 2 inches. It may be obtained in 50-foot lengths. It is a good non-conductor of heat and has, also, the advantages of being slow to absorb moisture and easy to keep in place.

Unlike most of the other materials mentioned, hair felt is not a filler but is laid over the surface of the wall. To apply it to a brick-walled surface, a sheathing of matched and grooved boards is nailed over the studding or nailing strips on the wall, and this sheathing is then covered with a layer of waterproof paper. In this way, an air space is made. Furring strips are now run vertically from floor to ceiling, and between them are fitted strips of hair felt from 1 to 2 feet wide. One layer of felt is laid over another, until the insulation is as thick as desired. To finish the wall, another sheathing of waterproof paper and boards is fastened over the felt. Since the nails which held the felt between the furring must be taken out as this finishing layer is put on, they are not driven through the felt but are driven part way through the furring and bent over in such a way as to hold the felt.

The thermal conductivity for hair felt is about 0.246 B.t.u. per square foot per hour per inch of thickness per degree Fahrenheit. The density in pounds per cubic foot is 17.

Wood.—Dry wood is a good non-conductor of heat and is, besides, easy to supply in quantity and to work on. In some refrigeration work, layers of planks have been built up for a thickness of 6 inches or more; but for general use, it is more economical to employ wood in connection with some good filler. Several walls can be constructed, one within the other, and the air space between each two filled with insulation. Boards used in this way must be matched and fitted to form an airtight wall. Rough inside joints can be covered with waterproof paper.

Some woods possess the disadvantage of having a strong odor and of absorbing moisture too readily. Spruce, white pine, hemlock, and basswood are free from these objections and make good insulation, although spruce and basswood are hard to obtain. White pine which has been thoroughly seasoned and freed from rosin makes an insulation second only to spruce in quality. The splintery nature of hemlock makes it more vermin proof than the other woods, but, for the same reason, it is hard to cut and nail.

In general the thermal conductivity for wood is much higher than cork. The thermal conductivity varies from 0.25 B.t.u.

per square foot per hour per inch of thickness per degree Fahrenheit for balsa to 1.13 for hard maple.

Shavings.—Where shavings are packed into a large space, they do settle to some extent; but if the space between walls is divided into small compartments by nailing pieces between the studding, trouble of this kind is prevented. Of course, shavings are not absolutely moisture proof, and wherever they are used, care should be taken to protect them against dampness. The same precautions against odor must be taken with shavings as with wooden planks; spruce, hemlock, and white pine are again the best. For convenience, shavings are frequently compressed into bales. In this form, they are much easier to ship and to handle. Shavings should be kiln-dried.

Mineral Wool.—A fireproof, vermin-proof insulator with many fine air cells is obtained through the use of mineral wool. It is made by melting furnace slag and limestone and blowing the molten mixture into a fleecy mass by an air blast. The chief objection to this kind of insulator is its brittleness. The fibers break very easily if packed; and because the sharp ends prick the hands of the workmen and fine particles irritate their eyes, it is hard to get men who will do thorough work in using it as a filler. This disadvantage is overcome, to a certain extent, by manufacturing the wool in sheets of various sizes and thicknesses. Such sheets can be handled quite easily. Dampness attacks mineral wool very quickly and injures its insulating qualities. For this reason, mineral-wool filling should always be protected by some waterproof material.

The disagreeable features of mineral wool have been overcome so that today the workman can handle it better than formerly; the fibres are not as brittle having been annealed. The thermal conductivity is about 0.28 B.t.u. per square foot per hour per inch of thickness per degree Fahrenheit. The density is about 13 pounds per cubic foot.

Celotex.—Celotex is a form of insulating material which is made from cane fibers (bagasse). Like all other similar materials, it consists of minute air cells. It is not only used for refrigerating purposes but as wallboard for structural purposes as it has considerable strength. Celotex insulation can be purchased in shapes cut to the exact dimensions so that the cost of erection is low.

In manufacturing celotex the bagasse fibers are cooked to dissolve out all soluble matter and the fibers are, therefore, prac-

tically pure cellulose. In the process of manufacture, waterproofing chemicals are added. Celotex is odorless and light, weighing about 13 to 14 pounds per cubic foot and when properly applied will not settle. The thermal conductivity of celotex is 0.33 B.t.u. per square foot per hour per inch of thickness per degree Fahrenheit. Like all insulants, celotex should be sealed by means of suitable asphalts in order to maintain its low insulating property.

Dry Zero.—An insulating material called "dry zero" is made from fine tubular fibers which come from pods of Ceiba trees which grow in the tropics. The pods are gathered by natives and are sun-cured before being sent to this country. Here they are made into a heat-insulating felt. In graining the felt, the fibers are made to lie at right angles to the line of heat flow.

Dry Zero blanket is made of the grained felt enclosed between layers of rot-proof burlap and is stitched every 6 inches. It is made for refrigerating cars, aeroplanes, truck bodies, partitions, and walls. Dry zero "blanket" is made with a waterproof covering thus producing a heat insulator which may be especially used for truck bodies.

Dry Zero pliable slab is made from grained felt completely enclosed in a stiffened waterproof envelope, composed of plastic asphalt and tough paper. This is the form of dry zero which is used for refrigerators, cabinets, counters, and cold-storage doors.

The conductivity coefficient for heat transfer of dry zero, as determined by the U. S. Bureau of Standards is 0.24 B.t.u. per square foot per hour per inch of thickness per degree Fahrenheit.

Sawdust.—Sawdust serves as an excellent covering for stored ice, but it is not one of the best insulators for filling the walls of cold-storage compartments. It has a high insulating value only when it is perfectly dry; wet sawdust conducts heat rather quickly. Since sawdust is usually obtained from green wood, it is difficult to get in a dry condition, and even when it is packed dry, it absorbs moisture unless carefully protected.

If sawdust is packed in air spaces while damp, it tends to settle when it dries out, thus leaving an unprotected space at the upper part of the wall. When wet, it rots or ferments and gives off an odor which is likely to taint cold-storage goods. Another objection to sawdust is that it attracts vermin and furnishes a good nesting place for rats and mice.

Straw and Chaff.—Chaff, hay, straw, and grass chopped into fine pieces are sometimes used for insulation, but they do not belong to the better class of insulating materials. Like sawdust, they are excellent non-conductors of heat when dry but are very likely to absorb moisture and to rot.

Charcoal.—Charcoal is used as insulation in some European cold-storage plants, but its use is not favored in this country. While it is an excellent non-conductor of heat, it absorbs moisture, is dirty to handle, and costs more than some better insulators.

Insulating Papers.—All of the insulators which have been taken up have been used for the purpose of preventing the passage of heat. Insulating papers are intended rather to protect other insulating materials from moisture and warm air. They are also fairly useful as non-conductors of heat. The best insulating papers are those that are heavy and that have been coated with asphalt to give them greater durability. Tarred or oiled paper is to be avoided on account of odor, and paper sized with rosin is likely to disintegrate.

In applying the paper, care must be taken not to tear or puncture the sheet, for paper damaged in this way will permit the passage of air. As few nails as possible should be used. It is advisable to use several reinforcing layers on corners and at points likely to be exposed to hard wear.

Cardboard Cartons.—The use of paper has not been commonly considered a satisfactory material to use as heat insulator, but with the advent of frozen foods cardboard cartons have been successfully used. A form of paper insulation has been developed for insulating refrigerator cabinets. In the type of carton, shown in Fig. 235, the combined insulating effect of the liners and the carton is about the same as that obtained with 1 inch of cork board. Frozen foods that have been shipped in these cartons, after several days without other refrigeration, were received in good condition.

Satisfactory results have been obtained by using paper having a spacing of approximately 1/10 inch for flat sheets and about three corrugations to the inch for the corrugated separators. The heat conductivity for this arrangement is about 0.27 B.t.u. per square foot per hour per degree Fahrenheit per inch of thickness.

Paper does not deteriorate rapidly with age and, therefore, insulating paper made from *unbleached-sulphate* paper should

have a life comparable to rag-stock paper. It is known that light, heat, and air cause paper to deteriorate, but these factors are greatly reduced when the paper is used in a refrigerator cabinet. All paper insulation must be protected against moisture and one method used is to encase the insulating sheets in a suitable carton which is dipped in a bath of melted wax.

Quilting.—Quilting is made by placing layers of insulating material, such as mineral wool, flax fiber, hair felt, or seaweed, between two thicknesses of insulating paper and stitching the whole sheet together.

Flax fiber is made from flax straw which has been crushed and treated to deodorize it and remove the nap. It is best used in quilting, for, in that way, it has the protection of waterproof

FIG. 235.—Insulated paper cartons for frozen foods.

paper against dampness. There are several advantages in the use of eel-grass or seaweed quilting. Such quiltings are good non-conductors of heat and are extremely durable. Seaweed contains a great deal of iodine, and, for this reason, rats and mice avoid it.

Aluminum Foil.—This type of insulation was first investigated in Germany by Dr. Ernst Schmidt and is being sold in this country under the trade name of alfol. It is made by forming successive layers of sheet-aluminum foil approximately 0.0003 inch in thickness, spaced approximately ⅓ inch apart. The spacing may be made by corrugating the sheet material. A simple way of spacing is produced by crumpling the sheet aluminum so as to produce irregular ridges and valleys throughout the sheet. In this way the foil produces its own spacing with a minimum number of contacts. The length of the foil when properly prepared is reduced by crumpling about 10 to 15 per cent. The

amount of heat transferred by conduction through the aluminum foil is small because of its thinness and the limited contact area of the foil.

Polished aluminum has a high reflectivity for radiant heat; and this property is not affected by long exposure to the atmosphere at high temperatures. Highly polished aluminum reflects about 95 per cent of the radiant heat that strikes its surface. Copper and silver reflect about 96 per cent of the radiant heat but do not retain their polish. The surface of aluminum foil is covered with a layer of oxide that is transparent and protects the metal from being attacked by impurities in the atmosphere. It is stated that after 3 years there is no change in the reflectivity of aluminum foil.

A comparison of the heat-transfer coefficients for crumpled alfol, plain alfol, cork, and 95 per cent magnesia is shown in Fig. 236. The curves show that plain aluminum foil is superior to the other insulating materials compared. Aluminum-foil insulation as ordinarily used weighs about 3 ounces per cubic foot. When aluminum foil is used in place of cork for household refrigerators, the weight is reduced about 35 per cent, and there is also a 5 per cent reduction in heat loss.

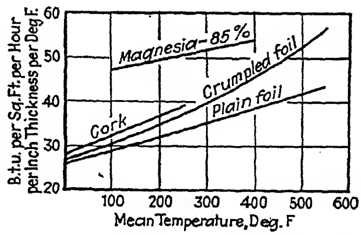


FIG. 236.—Heat-transfer coefficients for alfol, cork, and magnesia.

This type of insulation may be used for sub-zero temperatures as well as up to 1200° F. which is the melting point of aluminum. It has been used for insulation on brine and steam piping, ice-cream containers, refrigerated rooms for storing fish and meat, and household refrigerators. Aluminum foil is fire- and vermin-proof and has high resistance to industrial fumes and vapors.

Heat Transfer through Insulation.—Nearly all of the insulated walls used in cold-storage construction have been tested for the rate of heat transfer between their surfaces. With this information, it is possible to calculate the amount of heat passing through a heat-insulating wall. Such calculations are not accurate in all cases, because heat-insulating materials vary in condition and are constructed under variable conditions.

The values obtained from such tests are expressed in the number of B.t.u. per hour passing through 1 square foot of wall surface

for each Fahrenheit degree difference in temperature between the two sides of the wall. This value is called the *heat-transfer coefficient* for the wall. The amount of heat transferred through the wall can easily be calculated if we know the number of square feet of wall surface, the heat transfer coefficient, and the difference in temperature between the two sides of the wall. The amount of heat passing through the wall in 24 hours can easily be found by multiplying the value found above by 24. This may be expressed in a single formula.

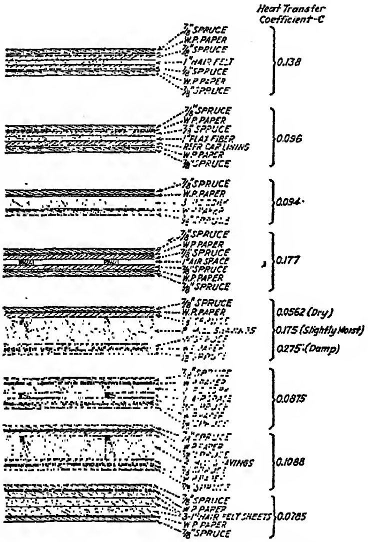


FIG. 237.—Heat-transfer coefficients for cold-storage construction.

Heat passing through wall per hour $= AC(t_2 - t_1)$, where A = number of square feet of wall surface; C = heat-transfer coefficient; t_2 = temperature of outer wall; t_1 = temperature of inner wall.

As an example, suppose the area of a wall to be 1,500 square feet, the heat-transfer coefficient to be 0.11, the outside temperature 80° F., and the inside temperature, 20° F. Then A = 1,500 and C = 0.11; $(t_2 - t_1) = 80 - 20 = 60°$ F.

The heat transferred per hour $= 1{,}500 \times 0.11 \times 60 = 9{,}900$ B.t.u. per hour. For 24 hours, the amount will be $9{,}900 \times 24 =$ 237,600 B.t.u

necessary 237,600/288,000 or 0.824 ton refrigerating capacity.

This means that additional refrigerating capacity of 0.824 ton is required to remove this heat. In case the wall had a heat-transfer coefficient twice the one used, the heat passing through would be twice as great. This shows the importance of constructing the walls so as to have a low heat-transfer coefficient.

Some cold-storage compartments have windows which allow considerable heat to pass. The amount of heat transferred in this way can be found by the above method.

The heat-transfer coefficient for a single thickness of window glass is 1.0; for double windows with air space between, it is 0.45.

Figure 237 shows some of the methods used in constructing cold-storage walls. It also gives the heat-transfer coefficient for each type of wall.

Heat Principles.—Heat is the result of the violent motion of molecules or particles which go to make up a substance. The transfer of heat is a matter of first interest in the study of refrigeration. It takes place in three distinct ways: (1) convection, (2) conduction, and (3) radiation.

Convection is the simplest to understand. The word itself comes from a Latin one meaning "to carry." That is exactly what happens: the heat is *carried* from one place to another by some fluid, such as air or water, as when a chip of wood is dropped into a running brook, the chip is carried away by the motion of the water. Now, suppose that some hot water is poured into the stream. At once it enters into circulation with the other water and flows downstream. If a thermometer is placed in the current a few feet downstream, it will register an increase in temperature. In other words, heat is transferred or carried from one place to another by the circulation of a fluid. In a common type of cold-storage room, warm air rises to the top of the room, is cooled by contact with the refrigerating coils, and falls again to the floor. Thus, heat is being transferred *from* the stored goods *to* the refrigerating coils by the circulation of a fluid—air.

Conduction is the transfer of heat *through* a substance. A familiar illustration is the silver spoon in a cup of hot coffee. The bowl of the spoon rests in the coffee; the handle is outside—in the air. In a few moments, the handle of the spoon, though it has not touched the coffee, will be hot. The heat has passed up through the spoon into the handle. To understand thoroughly how this takes place, the theory of heat must be kept in mind. The silver spoon is made up of particles or molecules. The coffee, coming into contact with the bowl of the spoon, sets the molecules violently in motion. These molecules knock against their neighbors. The process goes on until all the molecules in the spoon are agitated. In a game of billiards, a player sets one ball in motion by striking it with a cue; that ball strikes another ball and sets it in motion; and so on. It might be imagined that each ball is a molecule. The transfer of motion, in that instance, corresponds to the transfer of heat by conduction.

The third method of heat transfer is *radiation*. Less is known about radiation than about either conduction or convection, but it is clear that a third form of heat transfer exists, dependent neither upon the circulation of a fluid nor upon molecular activity within a substance. For instance, if a stove poker is heated so that it becomes red hot and a hand is then placed a few inches beneath the heated end, there is a sensation of heat. Since heated air rises, obviously, the air heated by the poker goes away from the hand instead of toward it. The heat of the poker is not transferred, therefore, to the hand by convection. Air can conduct heat, but the process is so slow that it takes a considerable time for heat to travel by conduction. The transfer is, then, by the third method, which we call *radiation*. It is the one way by which heat can pass through a vacuum, and, for this reason, it is supposed to depend on vibrations of the ether.

When a cold-storage room is insulated, all of the three kinds of heat transfer are encountered. Some insulating materials are selected because they conduct heat very slowly; still air is much used because it prevents convection; and screens or layers of various materials are employed to guard against the direct radiation of heat. But in constructing walls for a cold-storage plant or a cold-storage compartment, granulated cork, mineral wool, and the other heat-insulating substances are not sufficient: wood, brick, or stone masonry is needed to give support, especially in the outer walls of a building; and, in some cases, partitions are necessary to hold the real insulator in place. Of course, such materials as brick are used principally for strength, rather than for any insulating qualities; but they have, nevertheless, the power to retard heat, to some extent.

In composite walls (insulated lining combined with the outer walls), the coefficient of heat transfer cannot be based on the insulated lining alone but must take into account the passage of heat through the outer walls.

In order to determine the heat-transfer coefficient for a *composite wall*, the surface coefficients, thermal conductivity for the various materials, and the thickness must be known.

In the case of a solid composite wall the heat-transfer coefficient, C, may be calculated from the following equation,

$$C = \frac{1}{\frac{1}{a_1} + \frac{1}{a_2} + \frac{1}{a_3}\ etc. + \left(\frac{x_1}{c_1} + \frac{x_2}{c_2} + \frac{x_3}{c_3}\ etc.\right)}$$

where a_1, a_2, a_3, are the surface coefficients for the various materials (Table XIV, p. 522) in B.t.u. per square foot per hour per degree Fahrenheit difference, x_1, x_2, x_3, are the thicknesses of the materials in inches, and c_1, c_2, c_3 are the thermal conductivities (Table XVII, p. 523) for the various materials in B.t.u. per square foot per hour per degree Fahrenheit difference.

When a composite wall contains air spaces, the coefficients for still air and the surface coefficients for the various materials must be used. In case the air spaces are large, little insulating effect is obtained, because the circulating air transfers the heat from one surface to the other by convection.

The heat-transfer coefficient, for example, may be determined as follows for a wall which is made up of 13 inches of brick, ½ inch of cement, 4 inches of corkboard, and ½ inch of cement plaster to finish the surface of the corkboard on the inside wall.

From Tables XVII and XVIII in the appendix the thermal-conductivity coefficients may be obtained and are as follows: brick, $c = 5$; corkboard, $c = 0.3$. The coefficient c for plaster may be taken the same as for brick, that is, $c = 5$. The surface coefficients for plaster and brick may be taken from Table XIV (p. 522) $a_1 = 1.1$ and $a_2 = 1.4 \times 3 = 4.2$ respectively. It should be noted that the effect of moving air on the outside surface of the wall has the effect of increasing this value for still air in some cases as much as three times. The heat-transfer coefficient for this composite wall can then be calculated as below,

$$C \qquad 1.1\bar{0} + 4.2 \qquad \frac{4}{0.3}$$

$=$ 0.058 B.t.u. per square foot per hour per degree Fahrenheit difference.

Air Infiltration.—It has been observed that some cold-storage rooms which were constructed with the usual thickness of cork board for insulation did not give satisfaction when compared with the expected results. Recent investigations show that the warm air that leaks into a room and is there cooled below its dew point deposits its moisture in the insulation so as to raise its heat conductivity; and, at the same time, it weakens the walls. When still air is cooled in a refrigerated room, the cooled air near the ceiling has a tendency to flow toward the floor and thus create a negative air pressure at the top of the

room and a positive air pressure at the bottom. There is, therefore, a flow of so-called infiltrated air into the room at the top to equalize the negative pressure there, and cold air is forced out at the bottom due to the positive pressure. The quantity of air infiltration will be affected by the velocity of the wind and will be greatest on the side of the building toward the prevailing wind. With solid brick walls 24 to 48 inches in thickness, as commonly constructed many years ago, there was an entirely different type of wall in regard to heat insulation in comparison with the light "curtain" wall as constructed today. Little attention is given to the air-tightness of the ordinary brick wall as now constructed; and, consequently, about 20 per cent of the wall area is mortar which offers but little resistance to air flow through it. In fact, a curtain wall of reinforced concrete has greater resistance to air infiltration than a brick wall of the curtain type. The results of tests show that at a pressure of 40 pounds per square foot, which is considered as equivalent to a great hurricane, the air infiltration is 115 cubic feet per square foot per hour for an ordinary 13-inch brick wall with a ½-inch coating of portland cement on the inside. This amount of air if allowed to pass through the insulation of a building will deposit moisture in the insulation long before the frost line is reached. The variation of temperature on both sides of the insulation causes the frost line to shift, which eventually produces disintegration of the heat-insulating material.

Recently it has been demonstrated that two coats of good asphalt paint sprayed on the insulation will almost entirely prevent air infiltration. In the case of corkboard having a thickness of 2 inches covered with a ⅛-inch film of erection asphalt with no cracks, the air infiltration was zero with an air pressure of 40 pounds per square foot of area.

Tests have been made to determine the relative infiltration through various thicknesses of corkboard. The density of

Thickness in Inches	Infiltration, Cubic Feet per Square Foot
1	485
2	343
4	186
6	150
8	124
10	100
12	95

the corkboard used in these tests was 0.854 pound per board foot.[1] The infiltration is given in cubic feet per hour per square foot of surface area and at a pressure of 40 pounds per square foot.

Since all air other than air having a relative humidity (p. 453) of zero per cent contains some water vapor, it follows that the air entering the insulation by infiltration will carry its moisture with it. The amount of water in the form of vapor varies with the temperature of the air and its relative humidity. Air

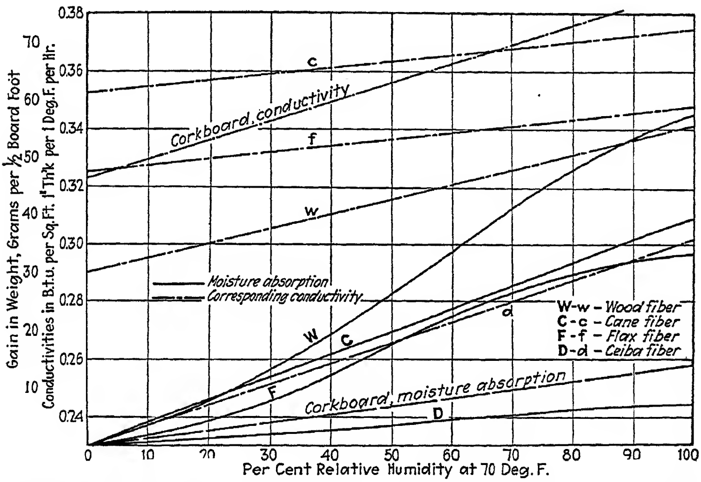


FIG. 238.—Heat-transfer coefficients and moisture absorption of insulating fibers and corkboard.

having a temperature of 75° F. and a relative humidity of 60 per cent contains 5.6 grains of moisture per cubic foot (p. 455). If this air is cooled to the dew point which is 59.6° F., the air becomes saturated with water vapor. If then the temperature is lowered to 40° F. the maximum amount of water vapor that the air can contain becomes 2.8 grains per cubic foot; and the difference between 5.6 and 2.8 grains or 2.8 grains is the amount of moisture per cubic foot of air that is condensed and deposited in the insulation.

The change in conductivity and moisture absorption at various relative humidities has been studied by Prof. L. P. Miller of the

[1] A *board foot* is a unit of measure for lumber. It is 1 square foot in surface area and 1 inch thick.

University of Minnesota. The results of these tests are shown graphically in Fig. 238. The tests were made on corkboard, wood, cane, flax, and ceiba fibers (dry zero, p. 368).

Masonry Priming Paints.—In erecting corkboard against masonry walls in asphalt, the kind of priming paint used is of considerable importance. It is important that the walls are dry, and the paint is thin enough to enter into the pores. A spray gun will give a greater penetration of the paint than can be obtained with a brush. In the case of a small job, where the spray gun would not be feasible, a good finish, however, can be obtained with a brushing. In general, painting with a brush is better than "mopping" with hot asphalt. The solvent in paint decreases the viscosity and allows the paint to enter more readily the pores of the wall surface. After the solvent evaporates, the asphalt remains embedded in the pores. If the walls are damp the asphalt will not be well bonded.

In addition to the above requirements, the film of asphalt must not soften when the sun shines on the wall. Often temperatures as high as 140° F. on the outside of the wall will produce a temperature of the paint of about 105° F. A good paint may be produced by combining a good waterproofing asphalt with stearine pitch which is an oxidizable material. The stearine pitch absorbs oxygen and dries like linseed oil forming a hard finish. The melting point of the mixture before it is oxidized is 140° F. The oxidation of the pitch increases the melting point and the temperature at which the film will flow.

Cold-storage Buildings.—The use of insulating materials with a low heat-transfer coefficient is not sufficient to obtain efficient cold-storage effects. The kind of construction of the building itself is equally important. The first rule is that all the materials used should be of high quality. This does not always mean that they should be the most expensive on the market; but it does mean that the purchase of inferior materials is certain to be poor economy in the end. The two results to be looked for are, of course, good insulating capacity and durability. So far as durability is concerned, the same principles hold good in cold-storage construction as in any other form of building. Materials which will not warp, settle, or decay, and good workmanship in putting them together will avoid the expense of constant repair and the possibility of leakage on account of joints which are not tight. Because of the difficulty of renewing material which has

been built into the walls, it is essential, too, that substances be chosen which will not only retard the passage of heat when new but which will also retain their insulating properties. The building should be arranged with a minimum number of doors and windows. No skill of the architect or of the builder can do away with a certain amount of refrigeration loss through and around doors and through the windows. This is due, to some extent, to doors which do not fit snugly. It is desirable to buy special insulated doors which are manufactured especially for cold-storage buildings. These give better satisfaction than ordinary doors.

With windows, the case is different. Windows cannot be constructed without glass, and glass transmits heat more rapidly than any of the other ordinary materials used in a building. The best plan is to do without windows as far as possible. Sufficient light can be easily provided by a well-planned electric system of lighting. Electric lights give off some heat, but the refrigeration loss is much less in proportion to the illumination than it would be by the use of windows. And it must be remembered, at the same time, that the heat from a modern tungsten bulb ceases as soon as the light is turned off, while windows are a constant source of heat leakage.

Heat from Electric Lights, Motors, and Workmen.—In calculations for cold-storage rooms allowance must be made for the heat developed by lights, machinery, and workmen. The following table gives the amount of heat in B.t.u. per hour for each electric light during the time that it is lighted. In the table the power required for electric lights is in watts.

Capacity of Electric Lights, Watts	Heat B.t.u. per Hour per Electric Light
25	85.25
50	170.50
100	341.00
200	682.00
400	1,364.00
600	2,046.00

The table was calculated on the basis of 1 horsepower being equivalent to 746 watts and also equivalent to 2,545 B.t.u. per hour.[1]

[1] It is customary to allow one-half to 1 watt per square foot of floor area in cold-storage rooms for the amount of heat which is on the average given off by electric lights.

The heat generated by electric motors, fans and other machines is calculated in very much the same way as for electric lights, knowing that 1 horsepower is equivalent to 2,545 B.t.u. per hour or 42.42 B.t.u. per minute. In the case of electric fans used for air circulation all the electric power used by the motor driving the fan develops heat in the cold-storage rooms, so that the total amount of power actually used by the motor is converted into heat. In the case of motors used for operating conveyors and lifts the total horsepower actually used by the motor is converted into heat in the cold-storage rooms if no part of the power of the motor is used to operate machinery outside the cold-storage rooms.

The heat given off by men working in cold-storage rooms will vary from 400 to 600 B.t.u. per hour, depending on the kind of work. It is customary to assume that the heat introduced by a man when working is on the average 500 B.t.u. per hour.

The data given in this section will give the necessary information for calculating a large part of the amount of heat, aside from that entering through the walls, for which refrigeration must be provided. It is obvious of course that unnecessary heat should be prevented from entering the cold-storage rooms since all this must be removed by the refrigerating system.

Design of Cold-storage Rooms.—All the preceding calculations in heat transfer have been made on the basis of B.t.u. *per square foot of wall surface*, understanding that a rectangular room has six walls—that is, four side walls, a ceiling, and a floor. The floor and ceiling are included, because heat is as likely to pass through them as through the side walls. Since the amount of heat passing into a compartment with any given thickness of wall and efficiency of insulation depends on the number of square feet of wall surface, the best results are obtained with the fewest possible number of square feet of surface. It may seem, at first, that this will depend on the cubic capacity of the compartment and that, in cutting down the wall surface, it is necessary to give up a corresponding amount of storage space, but this is not necessarily the case. For example, a room which has a cubical shape (that is, a room whose height, length, and breadth are the same) has the least wall surface for any given storage capacity. A room 10 feet high, 10 feet long, and 10 feet wide has a volume of $10 \times 10 \times 10$, or 1,000 cubic feet.

Now, to find the total wall surface, the area of the surface of each wall must be calculated, and then the surfaces added

together. Each side wall, being 10 feet long and 10 feet high, contains 100 square feet; and four of these walls contain 400 square feet. The floor is 10 feet long and 10 feet wide and contains 100 square feet. The ceiling has the same dimensions and, also, contains 100 square feet. Adding the areas of the floor and ceiling to the area of the side walls, the total wall surface is 600 square feet.

With the same height for the room as before, suppose, now, that the other dimensions are changed so that the room is 25 feet long and 4 feet wide. The volume of the room will be 25 × 4 × 10 or 1,000 cubic feet, exactly the same as in the first example. Two of the side walls—the long ones—are each 25 feet long and 10 feet high, containing 250 square feet each or 500 square feet for the first two sides. The shorter walls are 4 by 10 feet, containing 40 square feet each or 80 square feet for the two sides. The floor is 25 feet long and 4 feet wide, containing 100 square feet. The ceiling has the same dimensions as the floor and, also, contains 100 square feet. Adding all six surfaces together, the total wall surface is 780 square feet, which is 180 square feet more than the room of the same volume which has the cubical shape. The greater wall surface has not gained additional storage space, while it has greatly added to the amount of heat that can pass into the room in a given time.

Fireproof Construction.—Cold-storage work does not, in general, require fireproof construction. Where a cold-storage plant is located in a thickly built city section, it is necessary, of course, to have a building which conforms to the standards for such localities, but that is for the protection of the city, not of the storage house itself. The principal disadvantage of fireproofing is that the materials used in fireproof construction are usually poor insulators. A second disadvantage is the cost, which more than equals the saving from the lesser fire risk.

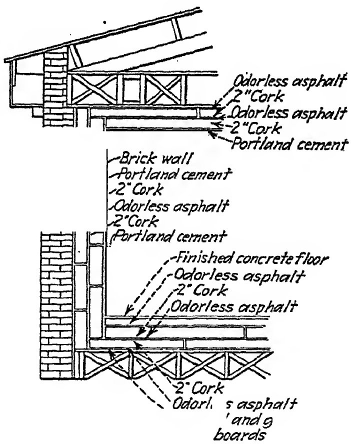


FIG. 239.—Cross-section of wall of modern cold-storage building.

Experience has demonstrated that few fires start in the cold-storage compartments themselves. Thus, a reasonable protection is given by the use of brick outer walls with heavy wooden construction within, preferably with a layer of smooth cement plaster inside the insulation, to lend fireproofing properties to the construction.

Wall-construction Design.—The cold-storage wall design shown in Fig. 239 is the cross-section of a modern cold-storage building.

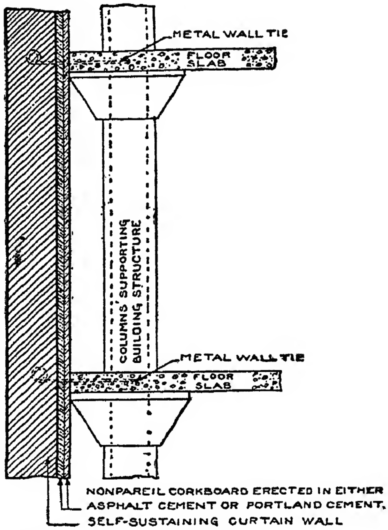


FIG. 240.—Self-sustaining curtain wall in cold-storage construction.

Cold-storage Insulation.—In insulating a cold-storage room, the engineer tries to make it an island in an ocean of heat. Many substances which have been used for insulation have been taken up. Of these substances, cork is most commonly used today. Cork is suitable for cold-storage insulation because it is: (1) a good non-conductor of heat; (2) moisture proof; (3) durable; (4) odorless and sanitary; (5) slow burning and fire retarding.

A modern cold-storage house is built of fireproof material throughout, being generally a concrete structure. It may be of steel framework with reinforced concrete ceilings and hollow tile walls. It is desirable to make the insulation of the walls continuous. This can be brought about by constructing a

self-sustaining curtain wall (Fig. 240), which is independent of the interior structure except for the small metal ties. The insulation is applied against the inner surface of the curtain wall, which is a continuous sheet without a break from the basement to the roof. *Corkboard* is generally used for this purpose. Generally, about 4 inches of cork board are laid in hot asphalt, in such a way that all transverse joints are "broken." This thickness of insulation will pass about 2 B.t.u. per square foot of surface per 24 hours, per degree Fahrenheit difference in temperature. The total heat transmitted through an insulation is expressed in B.t.u. transmitted per square foot, per 24 hours per degree Fahrenheit difference. This value for Nonpareil cork, of 1 inch thickness is 7.9.[1] For waterproof lith board, another common insulator, the value is 8.4. The following table gives the total heat passed per square foot of surface, per day, per degree Fahrenheit difference for various thicknesses of cork board.

Thickness of Cork Board, Inches[1]	Transmission, B.t.u. per Square Foot per 24 Hours per Degree Fahrenheit Difference
1	10
2	5
3	3½
4	3
5	2½

[1] For granulated cork of the same thickness, these values should be approximately doubled.

Generally speaking, it may be said that the thicknesses of the best cork board which can be economically installed for the several temperatures are as follows:

Temperatures, Degrees Fahrenheit	Thickness of Cork Board, Inches
−20 to −5	8
− 5 to +5	6
5 to 20	5
20 to 35	4
35 to 45	3
45 and above	2

The conductivity of various insulating materials is given in Table XVII, p. 523 in the Appendix.

[1] See "Heat Transmission," *Penn. State Coll.*, *Bull.* 30.

CHAPTER XI

SMALL COMMERCIAL REFRIGERATORS

Design of Small Commercial Refrigerators.—Since the development of satisfactory small refrigerating equipment, uses have been found for small commercial refrigerating systems

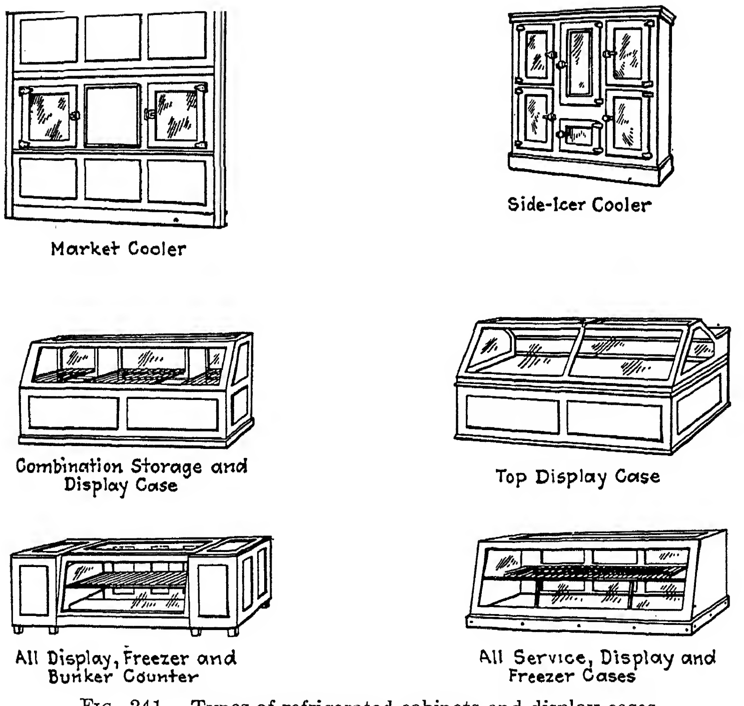


FIG. 241.—Types of refrigerated cabinets and display cases.

in hospitals, factories, schools, clubs, office buildings, restaurants, florist shops, meat markets, and drug stores.

In general, the type of condensing unit, consisting of a compressor, air- or water-cooled condenser, electric motor, fan, and liquid receiver, is designed like the household units (p. 135) but larger in capacity.

The refrigerator cabinets are built in various styles, as shown in Fig. 241, depending on the application. The type of evaporating coil used in these cabinets depends on the shape of the space in which the coil is to be located and the method of baffling.

Location of the Coil and Baffles.—Satisfactory operation of these refrigerators depends on the method of baffling, regardless of the size of the coil or the condensing-unit capacity. As heat is absorbed from the air in contact with the cooling coil, it is cooled, and becoming heavier makes a downward air current through the products to be cooled, from which it takes heat and then rises, thus coming again in contact with the cooling coil. To assist this circulation and to direct the flow of air, baffles are placed in suitable places. The air currents around a cooling coil and a baffle are shown in Fig. 242. It is not only necessary to have the air flow in a definite path, but it is also necessary to have the air passages or *flues* of the proper areas. If the warm-air flue is larger in area than the cold-air flue, eddy currents will occur and the air circulation will be poor. The same condition occurs if the cold-air flue is larger than the warm-air flue. On the other hand, if all the flues are too small the circulation will be checked and unsatisfactory refrigeration will result, because not enough heat is carried away from the products stored. Also if the flues are too large, the rate of air circulation is too great, and too much moisture is carried away from the products, thus causing drying and shrinkage. In the case of coolers up to 10 feet in width, the arrangement of the baffles may be as shown in Fig. 243, while for coolers greater than 10 feet in width the location of the baffles may be determined from Fig. 244. A drip shield and a drain trough are shown in Figs. 242, 243 and 244, which prevent falling water from the coil to drop into the storage compartment when the coil is defrosting (p. 152). The drain trough should be connected to an open drain, thus permitting the water to drain properly. The drip shield should only cover

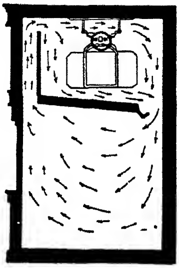


FIG. 242.—Air currents cooling coil and baffle.

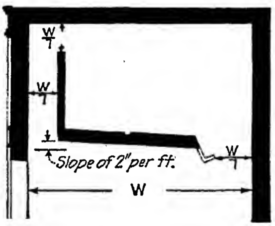


FIG. 243.—Arrangement of baffle in refrigerator less than ten feet wide.

the cold-air flue immediately under the flue and should not interfere with the circulation of cold air. In general, the upward current of warm air should be on the side where the doors are located. This is especially desirable in coolers made with doors that have glass panels, as it reduces the frosting of the glass as well as cuts down the loss of cold air.

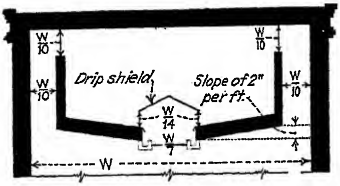


Fig. 244.—Arrangement of baffles in refrigerator more than ten feet wide.

In constructing baffles, care should be taken in selecting the kind of wood to be used, as odorous woods will affect foodstuffs. Spruce, fir, poplar, hemlock, birch, maple, oak, redwood, or elm may be used without danger of contaminating the foodstuffs. The wood should be painted with two coats of shellac before installing. A vertical baffle and a drip shield can be constructed of a layer of 1½ to 2 inches of cork board laid on spruce boards, over which, if desired, may be placed a layer of spruce boards. A trough will be required to carry away the defrosting water. This trough may be V or U shape and lined with galvanized iron. Likewise, the baffle should be covered with galvanized iron to protect the wooden surface and carry the water to the trough.

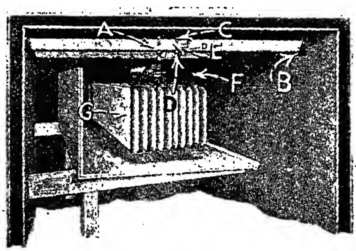


Fig. 245.—Method of suspending cooling coil over baffle in refrigerator.

The vertical baffle should be made of two thicknesses of ½-inch tongue-and-groove spruce, with a layer of odorless waterproof paper between the layers of wood. The distance between the top of the baffle and the ceiling of the cooler should be the same as the width of the warm-air duct.

The position of the cooling coil is quite important if efficient results are desired. The amount of heat absorbed by a cooling coil does not necessarily depend on the actual area of the cooling coil for a given difference of temperature between the coil and the cooler but on the effective coil area, which depends on the coil surface over which the free air is circulating. The cooling coil

should not be placed either too high or too low but about as shown in Fig. 242. In locating the coil it should be kept in mind that its fins should be parallel to the air circulation.

A satisfactory method of locating a large cooling coil is shown in Fig. 245. The coil is suspended by means of two 2- × 4-inch timbers, *A*, which are supported by four brackets. The coil is supported by means of two hangers, *F* and *G*, which are bolted to the cross timbers at *E*.

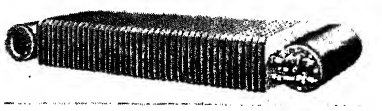


FIG. 246.—Cross-finned cooling coil.

The preceding information treats of the general methods of locating the cooling coil in the cabinet. When cross-finned coils, such as shown in Fig. 246, are to be used, however, a different arrangement is necessary. This type of coil is made for heavy duty and, therefore, must be kept free from frost in order to produce the required refrigeration, as the accumulation of ice on the fins restricts the air flow. This is not the case where the spacing between the fins is greater. The best location for this type of coil is shown in Fig. 247. This arrangement applies to overhead bunker coolers and other refrigerators. The coils must be installed high in the bunker to avoid eddy currents and sluggish air movements at the top of the boxes. The coil should be placed so that the top is 1 inch below the warm-air baffles. If there are two coils in the same bunker and one is adjacent to the cold-air flue, the top of this coil should be 2½ to 4½ inches below the top of the warm-air baffle.

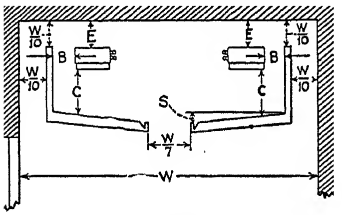


FIG. 247.—Location of heavy-duty cross-finned cooling coils over baffles.

Cooling Coils.—The cooling coils used in small commercial refrigerators are made to operate with either the dry or the flooded system (p. 63) and are made in various shapes and sizes depending on the shape of the cooling compartment and the height of the ceiling. It should be understood that the dry system is not intended to replace the flooded system but is used under conditions which would make the flooded system awkward or expensive to install.

The cooling coil for a typical top-display case resembles somewhat the one shown in Fig. 246. It consists of horizontal

tubes expanded and silver-soldered into drums, one being at each end. The larger drum at the right-hand end of the coil contains the float valve for maintaining a definite level of the liquid refrigerant. The left-hand drum is used to prevent oil clogging and adds to the efficiency of the coil. The fins are silver-soldered to the coils.

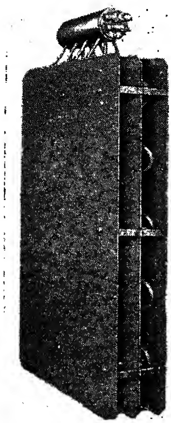


FIG. 248.—Plate or deep-finned cooling coil.

Another type of coil designed to be used in coolers and refrigerators which have ceilings too low for the use of overhead coils is the *plate coil* as shown in Fig. 248, which is also known as a "deep-fin" coil. The plate coil operates with the coil flooded with liquid refrigerant and has a drum located at the top which contains a float valve. The later types of plate coils have generally two large fins with tubes of the coil on each side of the fins. The earlier models had the coil located on only one side of the fins.

The plate-fin coil is generally located at the side of the refrigerator, as shown in Fig. 249. Due to the weight of the coil it is necessary to support it at the bottom by galvanized pipes and flanges. To tie the coil to the wall four Z-type braces are used. These braces are fastened to the coil by removing the nut from the tie bolt and placing the brace between the nut and the fin. These braces keep the coil at a uniform distance of 2 inches from the wall. The drip shield is set at an angle to deflect the air into the cooler. The top of the drum should be at least 2 to 6 inches below the ceiling, and the distance Y should be one-seventh of the distance W. The baffle should be located 2 inches away from the fin and built in two sections to permit servicing. The space indicated by V, between the bottom of the baffle and the drip shield, should be equal to one-seventh of the distance W. The baffle is generally constructed of 1 inch tongue-and-groove odorless wood with cork board 1½ inches thick between the boards. Considerable tem-

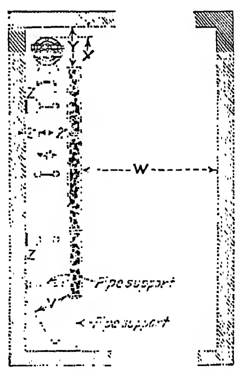


FIG. 249.—Location in refrigerator of plate or deep-finned cooling coil.

perature difference may be expected between the top and bottom of the cooler when using a plate coil. The maximum distance from the cooling coil at which good refrigeration can be expected is 7 feet. The depth of the coil should be about equal to the depth of the cooler.

Recommended Temperatures.—Table XV gives the recommended temperatures standardized by the Joint Refrigerating Committee of the Refrigerating Machinery Association and

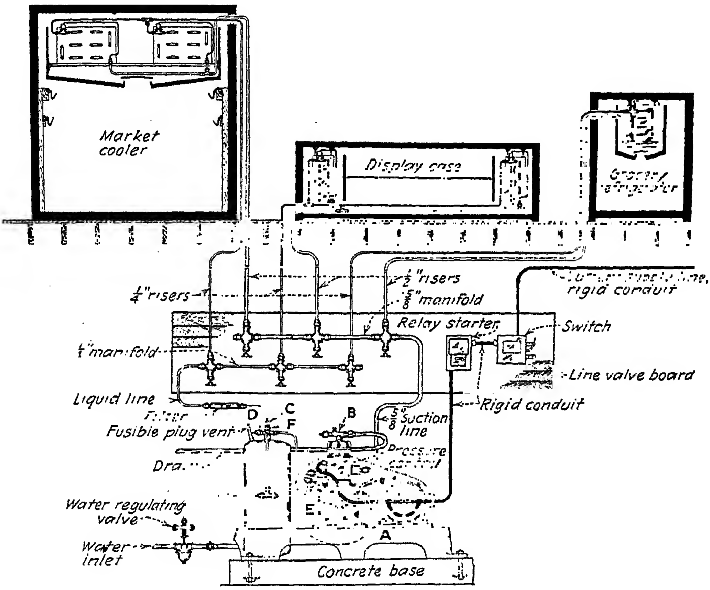


FIG. 250.—Typical market installation including refrigerated display case market cooler, and grocery refrigerator.

Commercial Refrigerator Manufacturers. This represents the findings of a competent group of investigators after exhaustive research.

Market Installation.—Figure 250 shows a typical market installation consisting of a display case, market cooler, and grocery refrigerator. These coolers are all equipped with fin coils and operate with the dry system. Each coil is controlled by thermal expansion valves. The arrangement of the piping for the high and low sides is clearly shown, as well as the electric connections.

TABLE XV.—TEMPERATURES RECOMMENDED FOR SMALL COMMERCIAL REFRIGERATORS

Normal or average operating conditions

Description of cabinet	Location of thermometer	From degrees Fahrenheit	To degrees Fahrenheit
Small market cooling room	Center of rear wall	38	45
Large storage cooling room	Center of rear wall	36	42
Grocer's refrigerator	Small lower compartment	42	48
Restaurant service refrigerator	Small lower compartment	42	48
Restaurant storage cooling room	Center of rear wall	38	45
Florist's refrigerator	Center of rear wall	48	54
Top-display case	Center of bottom	42	48
Floor-display counter	Center of bottom	42	48
Floor-display counter, heavy construction	Center of bottom	36	40
	Center of top-shelf	44	48

CHAPTER XII

AIR CIRCULATION AND VENTILATION IN COLD STORAGE

Air Circulation.—Some air contains more moisture than other air. Moisture in the air is called *humidity.* All air is naturally humid to a certain extent. Natural air does not exist anywhere in a perfectly dry condition. It is only when the percentage of moisture is relatively high, however, that we notice humidity.

Heated air absorbs more moisture than cool air; and air is less able to absorb or to hold moisture as its temperature falls. If moisture is present in sufficient quantity, air will finally absorb so much that it cannot hold more. It is then said to be *saturated.* At lower temperatures, it becomes saturated with less moisture than at higher temperatures. Thus, if air which was saturated at one temperature is cooled, it must give up some of its moisture by precipitation. The moisture given up in this way may remain in the air in the form of finely divided particles of water—as in fog—or it may be precipitated so rapidly that water gathers into large drops which fall as rain. For the same reason, when saturated air meets a cold surface, as the surface of a drinking glass, it gives up part of its moisture, which then gathers on the glass, so that the glass "sweats."

Saturated air when cooling gives up only as much moisture as the reduction in temperature makes necessary, and at the lower temperature it is still saturated; that is, it still holds as much moisture as possible *at that temperature.* Reversing the process, if cool air is heated, its humidity will diminish, for, at a higher temperature, it is capable of holding more moisture than it did originally. Humidity is, therefore, a matter of proportion, and the *relative humidity* of air is calculated by finding what percentage of the moisture which the air *can hold* is actually contained in it. Air when it is saturated at any particular temperature is 100 per cent humid at that temperature. Air which could hold twice as much moisture before reaching the saturation point has a humidity of 50 per cent.

It is necessary in cold-storage rooms to control the humidity of the air, for dampness interferes with the efficiency of the air for preserving goods. When dampness is permitted in cold-storage compartments, a musty odor soon develops that is injurious and objectionable. Dry air preserves perishable substances which damp air will mold. For these reasons, the air in a refrigerated compartment should be kept at a low humidity. Wherever goods—and especially meat—are stored, there is a good deal of moisture absorbed by the air from the goods.

One method of reducing the humidity in cold-storage rooms is to pass the air over some substance which will readily absorb the moisture. Some of the commonly used moisture absorbents are unslacked lime, common salt, and calcium chloride. Of these, the calcium chloride is most commonly used, because it is capable of absorbing a great amount of moisture in proportion to its own bulk and is inexpensive.

A second method depends on the effect of temperature changes on humidity. By this method, moist air is cooled until it becomes saturated, and then it is cooled further by contact with cold surfaces until it gives up some of its moisture. It is then heated so that it may again be able to hold more moisture, but since it has already deposited a certain amount of its moisture, it has less than it can hold, and its relative humidity is, therefore, less than it was before.

In some refrigerating plants, both drying methods are used at the same time. In either case, a circulation of air in the storage rooms is necessary; air cannot be cooled and heated without being put into circulation, nor can it be passed over absorbents without circulation.

Circulation is also needed to keep the air pure. The fungus germs which produce mold are abundant wherever goods are stored. If they are allowed to settle through stagnant air on the goods, they will develop rapidly and counteract the advantage of refrigeration. Circulating air will carry these germs away from the contents of the storage rooms and bring them into contact with the cold refrigerating pipes, where they will be deposited with the moisture. Since these germs cling to moist surfaces, they will be caught on the pipes and prevented from infecting the stored goods.

During the storage of foodstuffs, a gradual process of decomposition is constantly at work. It may be so slow in its action that

months will pass before the food is unfit for use. Certain fermentation gases, however, are always being formed, and, unless these are removed, they will lead to an unwholesome atmosphere in the storage rooms, as some of these gases are quickly absorbed by the air; they unite and move with the moisture in the air. Here, then, is another need for circulation. As circulating air deposits its moisture on the cold refrigerating surfaces, it deposits also the fermentation gases which would otherwise foul the storage compartment. Such gases as are not absorbed by the air may be cleared from the storage rooms by ventilation.

A third method of reducing the humidity in cold-storage rooms is by the circulation of air in order to maintain an even temperature in all parts of the storage compartments. If the air did not pass from one part of a storage compartment to another and were allowed to settle and remain stagnant, that part of the compartment next to the cold refrigerating pipes would have a temperature several degrees lower than that part in the middle of the room, so that goods stored near the sides of the room would be frozen before those in the center of the floor were cool enough to keep. Even if there were no danger of freezing goods near the pipes, there would be a tendency to put unnecessary work on the refrigerating machine in order to preserve goods at some distance from the refrigerating pipes. Circulation of air prevents such wastage by equalizing temperature in all parts of a storage room.

Methods of securing circulation are generally classed in two groups: (1) *natural* circulation, in which the air moves because of differences in densities due to changes in temperature; and (2) *forced* circulation, in which a fan generates and directs the air currents.

Refrigeration Effects of Ice-freezing Mixtures.—Though the melting of ice is a common method of refrigerating, there are several objections to this form of cooling. The first is the limitation of temperature at which ice can maintain a storage compartment. Ice melts at 32° F.; but this melting cannot keep a room at that temperature, except under the most favorable conditions. With summer heat to contend with, temperatures of 36 to 38° F. are possible. Most goods require storage at temperatures from 2 to 8° lower than this. If ice is mixed with common salt or calcium chloride, a compartment can be effectively cooled. Such a mixture melts at a temperature

lower than the melting point of unmixed ice; and because the difference between the temperature of the room and the melting point of the *mixture* is greater than between the temperature of the room and the melting point of *ice*, the mixture melts more rapidly and produces a greater cooling effect. If, for instance, a compartment contains air at a temperature of 33° F., the difference between that temperature and melting ice is 1° F. Now, because the rate at which the ice melts depends on that difference, the melting would be slow. If a sufficient amount of calcium chloride is mixed with the ice to make its melting point 20° F., the temperature difference between the calcium-chloride freezing mixture and the air in the room is then 13° F. Melting will then take place thirteen times as rapidly as when ice alone is used, and the cooling effectiveness of the operation would be greatly increased. A melting process is always taking heat out of the air and, thus, cooling the compartment.

Cooling with ice leaves the air too damp for the best results in cold storage. Air is most easily dried by bringing it into contact with surfaces which are so cold that the moisture it contains is condensed. The colder these surfaces are in relation to the temperature of the air the more moisture will be given up by the air. Now, in the ice-cooling methods of refrigeration, the ice-cooled surfaces are, at the most, but a few degrees colder than the air, so that the moisture which will be condensed is small. Thorough air drying depends on a wide temperature difference between the air and the cooling surface.

Besides temperature, there is one more condition that influences the humidity of air: the amount of moisture present which the air can absorb. Here, again, the ice-cooling method has a disadvantage, for melting ice forms water which the air will absorb as readily as it can. Natural air circulation, too, depends upon the range of temperature. Since the air of a storage room is nearly as cold as the ice, natural air circulation will be poor. In general, it is safe to say that ice-cooled compartments will not be satisfactory except where goods are stored for only a few weeks and where the required temperature is not below 38° F. Even to secure this result, some care must be given to the arrangement of an ice-cooled room. The earliest and simplest method was to place the ice directly in the storage compartment and to keep the goods at the lowest possible temperature. They were often placed in contact with the ice, as the average housewife

keeps a head of lettuce fresh. But this method did not keep the goods dry and frequently injured them. In any case, the placing of an ice bunker in the room caused the air to absorb a great deal of moisture and prevented satisfactory circulation. The obvious remedy was to put the ice bunker in a separate compartment, sometimes at the end and sometimes at the sides of the storage room. Because cool air settles and warm air rises, later arrangements were based on an ice bunker elevated above the storage floor and separated from it by a partition. Still more recently, racks were constructed near the ceiling of the room, and the ice was placed on them. The warm air of the compartment thus rose and flowed along the ceiling to the ice bunker, was cooled by

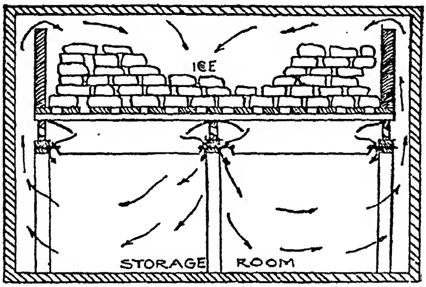


Fig. 251.—Air circulation in storage room refrigerated with ice.

contact with the ice, and fell again through the racks to cool the stored goods.

Pans of a dry absorbent, as, for example, calcium chloride, were so placed that the rising air passed over them and deposited some of its moisture. This method was an improvement, but it failed to provide good circulation of air or to reduce the dampness sufficiently; and the temperature of the storage room was still relatively high.

One of the best methods based on this system is shown in Fig. 251. Here a galvanized-iron dripping pan, placed under the ice racks, catches and drains away the water from the melting ice. Gaps are left around the sides of the pan to permit the passage of cooled air as it descends. The entire ice compartment forms an inner chamber within the storage room, and an air space is left between the walls of the room and the walls of the ice compartment. Through this air space, the warm air

rises and flows over the top of the ice. The cooling and falling of the air make fairly good circulation.

Piping Arrangements for Mechanical Refrigeration.—Most modern cold-storage plants require a refrigerating system of greater efficiency than can be obtained by the use of ice. For this reason, systems of mechanical refrigeration have come into use. In some of these, the refrigerant is expanded in the pipes; in others, cold brine is circulated. Either way involves the use of pipes and coils.

As the arrangement of ice bunkers affected the success of ice-cooled storage, so the placing of these pipes and coils is an important feature of mechanical refrigeration. The same three arrangements that were explained in connection with the use of ice bunkers hold good in regard to piping: The coils may be placed (1) in the storage room; (2) in a separate chamber divided from the storage room by a partition, or (3) above the storage room.

The best method of piping a cold-storage room or freezer cannot be definitely stated as the piping for each room is closely governed by the following conditions: (1) kind of goods to be stored; (2) when and how fast the goods are stored; (3) the method of storing goods; (4) size of room; (5) thickness and kind of insulation; (6) adjacent-room temperature and sun's radiation; (7) frequency or number of times the door is opened; (8) mean temperature desired; (9) humidity desired; (10) short or long period of storage.

The problem of low temperatures is very simply solved by the application of mechanical refrigeration and, when using ammonia as the refrigerant, temperatures as low as -60 to $-80°$ F. have been commercially obtained. When lower temperatures are needed, ethane or some other refrigerant may be used. The room temperature may be controlled by the pressure in the pipes which can be raised or lowered within limits; thus the outside temperature can be met, and an even temperature can be maintained in winter and summer.

To secure the best circulation, however, the arrangement of pipes must be taken into account. Clearly, natural circulation is possible when the coils are much lower in temperature than the air of the storage room, because, obviously, the circulation depends on the passage of air between the places of varying temperature.

The same difficulties of circulation and uniform temperature that appeared in the early forms of ice-cooled compartments are encountered when the coils are placed directly in the storage room. Goods in the center of the floor get insufficient cooling, while those near the pipes are in danger of freezing. But this objection can be overcome by erecting a thin wooden partition to prevent the direct radiation of heat from the goods to the coils.

The low temperature of the coils in systems of mechanical refrigeration is also advantageous in keeping the air dry. The pipes or cooling surfaces are so much lower in temperature than the circulating air that the air in striking them gives up much of its moisture, which immediately freezes on the cooling surfaces as frost.

Natural Circulation.—The special arrangements by which cooling by coils can be made most effective will now be discussed. Dryness of the air and uniformity of temperature depend very largely on good circulation of the air. In any special arrangements of refrigerating coils, therefore, circulation is of first importance.

The principle of circulation, already outlined, is this: warm air rises, and cold air falls. But it is necessary that a cold-storage room be so arranged that the rising air and the falling air shall not interfere with each other. That is the fault of a piping arrangement in which the pipes are distributed regularly over the ceiling. A body of warm air is always attempting to rise, but a body of cold air, occupying the same amount of space, is descending; the circulation, therefore, is complicated and ineffective. If, in such a cold-storage room, goods are piled high, those near the ceiling are liable to freeze. The plan of such a cold-storage room is shown on Fig. 252.

A better method is shown in Fig. 253. Here there are two groups of pipes, one on each side of the room, near the ceiling, and partitions to direct the currents of air under each group. These partitions are made up of shelving and are so arranged that the cold air, on leaving the pipes, will flow downward and outward toward the walls of the room. The warm air rises through the middle of the room between the inside edges of the partitions. Experiments have established certain proportions to be observed in building a storage room on this plan. The sloping partitions are generally inclined 6 inches in 10 feet; that is, on a slope of 1 in 20 or 5 per cent. The width of the air space

between inner and outer walls is one-twentieth of the width of the room.

Another important measurement is the distance between the floor and the bottom of the partition. This dimension varies with the width of the room. Where the room is wide, it should be low, so that the heated air will flow along toward the center of

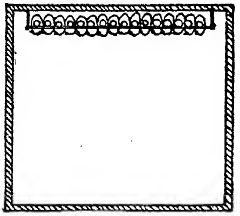


FIG. 252.—Cold-storage room with brine piping near ceiling.

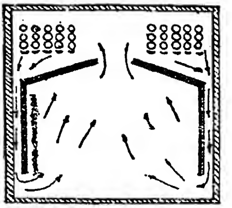


FIG. 253.—Cold-storage room with two groups of brine pipes

the room instead of rising too quickly and leaving a pocket of dead air in the middle of the floor. If the room is narrow, the height may be greater, for the reason that there is so little floor space to provide with circulation in proportion to the volume of the room that there is little danger of the air current rising too steeply.

A different arrangement based on the same principle is sometimes used for extremely narrow rooms. It allows for a single

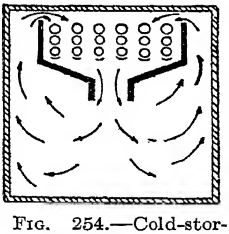


FIG. 254.—Cold-storage room with damp walls and floors.

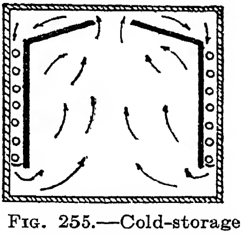


FIG. 255.—Cold-storage room with vertical rows of brine pipes along walls.

group of pipes at one side of the room, instead of the two groups as just explained. As a rule, the single-group system cannot produce such good results. Its value depends on economy in installation costs. As shown in Figs. 253, 254, and 255, the refrigerating pipes in each group are arranged in vertical rows, and under each row is fixed a long, narrow pan to catch and drain

off the drippings. Since there is a separate pan for each row, and a space between the pans, the cooled air is allowed to descend freely between the pipes. The use of one broad pan for all the rows would prevent this.

The wastage of floor space on account of the partitions and the difficulty of getting at the coils to make repairs are the two great disadvantages of the method shown in Fig. 253. The circulation secured by this method is, however, good, and because the heated, moisture-laden air gives up its humidity on the cooling surfaces before coming into contact with the floor and walls, these are kept in drier condition than with some other systems.

An example of damp floors and walls is shown in Fig. 254. The air currents absorb moisture from the produce and convey it to the walls. The goods themselves, however, receive cold, dry air. The disadvantages of this system are weaker and less even distribution of circulated air than in some other methods of air circulation.

Still another grouping and arrangement of pipes is shown in Fig. 255. The pipes are placed in vertical rows along the walls, and, unless the room is very narrow, a partition is built to direct the air current. The arrangements for drainage, in this case, are less complicated, and, on account of the simple side-wall construction, repairs to the coils can be made more easily than with the other methods. This arrangement, however, is not favorable to good air circulation, because the cooling surfaces of the refrigerating coils are not placed high enough. Clearly, a strong draft depends on having the coldest part of the room near the ceiling, so that the air, rising by the warmth that it has taken from the goods, will travel the whole height of the room before it becomes cool enough to fall again. In the method illustrated by Fig. 255, the warm air is forced to flow downward by the slope of the upper partitions and by the pressure of more warm air rising under it. When it has passed over the refrigerating pipes and has given up its heat, it flows along the floor until it is forced upward by warm air rising under it. When it has passed over the pipes and has given up its heat, it flows along the floor until forced upward by the rush of cold air behind or until it has slowly gathered warmth from the goods near the floor. Thus, instead of depending on the natural rising of warm air and the natural falling of cold air, this system depends on the forcing of warm

air downward (over the sloping partitions) and the forcing of cold air upward (from the floor). Either of the methods shown in Figs. 253 and 254 gives unusually good results in regard to air circulation because they are laid out with respect to the natural tendencies of air currents.

In general, it is best to plan the piping in bunkers or lofts with aprons, pans and baffles well insulated. This will produce a positive and rapid air circulation. When using wall coils it is often necessary to use an insulated apron to prevent the adjacent goods from freezing due to radiation.

The Cooper Combination Method.—Before the study of coil methods of cooling is dropped, the Cooper method deserves mention. It is neither distinctly a coil method nor distinctly an ice-cooling method but makes use of parts of both methods and combines some of the advantages of both. It was designed for use in those sections of the country where the supply of natural ice is plentiful. Crushed ice combined with calcium chloride is placed in a tank which contains a set of brine-filled coils. The brine is cooled by melting the ice, and, on cooling, it flows downward through a pipe to a secondary coil located in the storage room. There it accumulates heat and expands, rising through another pipe to the ordinary coils again. The circulation of brine is very similar to the gravity circulation of water in an ordinary hot-water heating system. Cool brine flows down by gravity, comes into contact with the heat given off by storage goods, warms, expands, and is forced to rise by the colder brine behind it. Pans of calcium chloride placed along the coils dry the air as it passes over them. By this method, the temperature of a storage room can be kept sufficiently low for any ordinary purpose. It is possible to maintain a room at 14° F. by this system, and rooms thus fitted have been refrigerated as low as 6° F. The value of such a system depends, of course, on a plentiful and cheap supply of ice.

Forced Air Circulation.—In all the cooling methods so far considered, the circulation of air has been produced by gravity. To secure good gravity circulation, that is, circulation caused by the rising of warm air and the falling of cold air, a storage room must have a special arrangement and grouping of refrigerating pipes, and, very often, partitions are needed, which occupy valuable storage space. On the other hand, several of the methods of natural or gravity circulation would be excellent if it

were not for the difficulties in obtaining at all times a satisfactory air circulation. For these reasons, most of the most up-to-date storage plants are equipped with fans to set in motion and direct the air currents.

By this mechanical means, a stronger and better circulation is obtained; the air is kept purer, and it is possible to maintain it in a drier condition. Another important advantage of forced circulation is that the temperature of the storage rooms is more even than it could be by the methods of natural or gravity air circulation. The gravity method of circulation depends on a relatively wide range of temperature in the storage room, so that warm air rises vigorously and the cold air falls in the same way. Sometimes, the difference in temperature needed to keep the air in circulation is enough to interfere with proper refrigeration. Air from different parts of a storage room provided only with natural or gravity air circulation will usually vary from 2 to 5° F.

Forced air circulation depends on no such uneven temperature. In fact, the strong currents generated by a fan so mix the air and drive it to every part of a storage room that the temperature is kept very even. The rapidity with which the air moves is an added advantage, for it speeds up all the processes for which circulating air is needed. This swift and well-controlled flow of air is of great assistance in keeping the air at a low humidity. The work of drying air in any system requires circulation over the drying surfaces, and any method providing a swift, positive flow insures a good control of humidity. Air purity is almost entirely dependent on two influences: (1) movement of air and (2) low humidity.

At different seasons of the year, the temperatures of the outside air varies; and this air, passing into the cold-storage rooms in numerous ways, affects the evenness of temperature inside. In the winter, this outside temperature may be lower than that of the cooling surface inside; in the summer, it is many degrees higher. In other words, the quantity of heat that may leak into the cold-storage rooms around the doors or windows or pass in through the walls or be carried in with the produce depends on the season. For this reason, the inside temperature is subject to seasonal differences. Now, the efficiency of natural or gravity air circulation, dependent as it is on the temperature difference between the cooling surfaces and the air, depends, also, on

seasonal change, to a large extent. When the temperature of the air is nearly the same as the temperature of the cooling surfaces, there is little or no gravity circulation.

With forced circulation, then, storage conditions, as far as circulation is concerned, are independent of weather and season. This advantage alone is enough to recommend the use of fan circulation wherever a large cold-storage plant is concerned. Although there has been much discussion as to the expense of forced or fan circulation, this system is, in fact, relatively inexpensive. There are but two important items of cost: (1) power for operating the fans and (2) cost of the air ducts. The power item is relatively low, 1 horsepower serving to supply 20,000 cubic feet of storage space. The cost of ducts cannot be figured in this way, but it is small compared to some of the costs of gravity circulation. For instance, the ducts occupy less room than the air passages in a natural or gravity system. In a comparison of costs, the additional space occupied by air passages, and thus taken from the room available for storage purposes, can be counted as the extra costs of the natural or gravity system. Another such cost is the extra refrigerating coil surface required where natural circulation is depended upon. Where the circulation of air is swift and strong, the air passes rapidly over the refrigerating coils and gives up its heat to them quickly. If the coefficient of heat transfer for direct-expansion piping is taken as 2 B.t.u. per square foot per hour per degree Fahrenheit (p. 248) for natural circulation, this coefficient will be raised considerably when air is forced over it by means of a fan. In fact, the coefficient may be as high as 5 to 10 depending on the velocity of the air through the coils. Forced circulation, then, accomplishes the same cooling effect with less refrigerating coil surface. Such items, balanced against the cost of installing a system of fans and air ducts, make clear the advantages of this method over the old, wherever cold storage is carried on on a large scale. And besides the question of economy, there are the advantages of simple operation, especially in the matter of drainage and defrosting.

Just as the problems of the gravity system have to do with the grouping and arrangement of the refrigerating coils, so the problems of a forced circulation system have to do with the placing of air ducts. The refrigerating coils themselves are usually placed in a heat-insulated bunker room at one end of the storage room. The suction side of the fan draws air from the storage

room, forces it to pass over the refrigerating coils, and blows it out through the cold-air ducts. These details may be varied, however, to suit the special requirements of any cold-storage plant.

There is a wide variety of arrangements for the cold-air ducts and the return ducts. One of the best methods provides for cold-air ducts along the floor, one at each side of the room. These ducts are long, boxlike passages with small openings along the top and sides. Through them the air is forced by the fan, and the pressure causes jets of air to spurt out at the openings. In this way, the cold air is thoroughly mixed with the air of the room. Upon absorbing heat, it rises to the return duct which is located in the middle of the ceiling. This duct, too, has openings in the bottom and sides so that it can receive the warm air from all parts of the cold-storage room.

For a narrow room, a cheaper arrangement is sometimes used, providing a cold-air duct along one wall at the floor and a return duct along the opposite wall at the ceiling. In this system, the circulation is diagonally upward; but, except where the room is small, the circulation will not reach to all corners of it. There are a number of other arrangements, each having its advantages.

One method which has given good results differs strongly from those already described. The storage room is fitted with a false floor and a false ceiling, each perforated with a large number of small holes. The cold air is forced through the false floor; and, on account of the small perforations, it enters the room in many fine air jets which mix thoroughly with the air already there. After absorbing heat, it passes from the room through the perforations in the ceiling.

Cold Diffuser.—Recently there has been developed a cooling unit which incorporates a direct-expansion or brine-cooling coil and a group of fans, which may be located in a storage room where a good circulation of cold air is necessary. These refrigerating positive air-circulating units are known as "cold diffusers."

The cold diffuser consists of a *flooded cooling coil* or brine coil as is shown in Fig. 256, and a fan assembly consisting of two, three, or four fans fully housed and driven directly on a common shaft by an electric motor which is mounted outside of the casing. The fans are of the centrifugal type properly balanced dynamically and statically. The shaft has self-aligning dust-proof bearings. The fans draw the warm air into the diffuser at

the bottom, and up through the closely nested cooling coil. These fans then drive the cooled air through directional outlets which control diffusion throughout the entire storage room.

The cooling coil is constructed of ¾-inch steel pipe with welded joints and headers. The entire assembly is hot-dipped galvanized. These cold diffusers are capable of handling a large variation of load, which is made possible by changing the motor speed. The capacities range from 1 to 15 tons of refrigera-

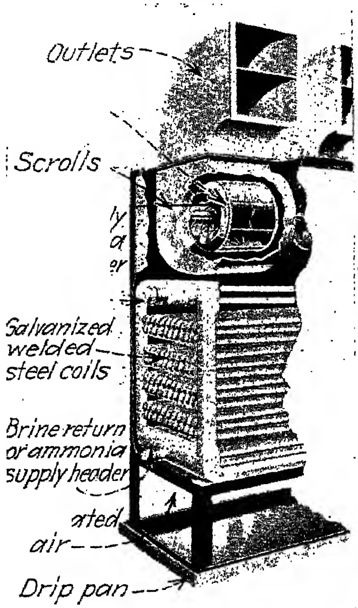


FIG. 256.—Simple type of cold diffuser.

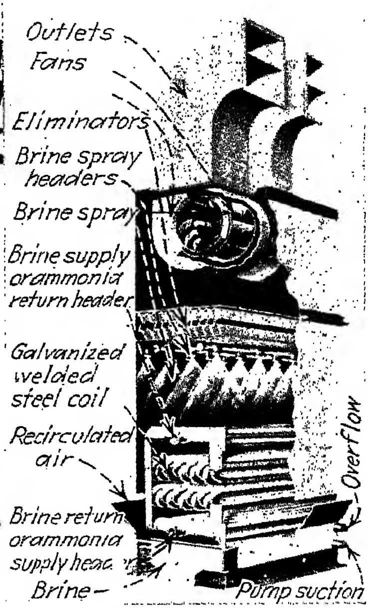


FIG. 257.—Cold diffuser with brine spraying device.

tion. The diffusers are suitable for holding rooms at a temperature of 33° F. and higher. This type is used where there is not a large amount of moisture given off by the product placed in storage.

Condensation of moisture on the walls and ceiling is eliminated by the positive-air circulation. The moisture in the air is condensed on the cooling-coil surface and drops into a drip pan at the bottom of the cold diffuser. The cold diffuser is designed for automatic or intermittent defrosting.

These units are designed to operate at higher suction pressures, which increases the refrigerating capacity of the entire system with lower operating costs.

Another type of cold diffuser, known as the brine *spray* type and suitable for holding rooms at 33° F. or higher, is shown in Fig. 257. This cold diffuser is built for use in rooms where excessive moisture is given off by the products in storage. In general, it closely resembles the above unit except that in addition to the cooling coil there are brine spray headers located above the cooling coil. Nozzles are provided for spraying brine over the cooling coil. The brine spray prevents frost from forming on the coil. A pump and motor, mounted on the

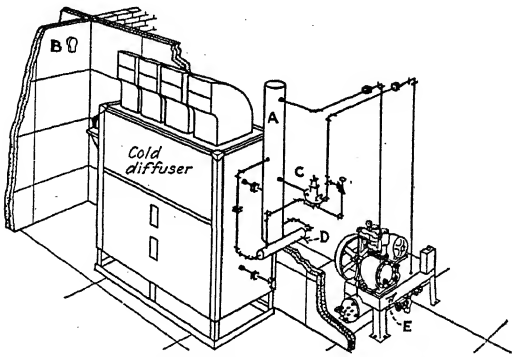


Fig. 258.—Cold diffuser with refrigerating unit.

unit, draws the brine from the tanks and discharges it through the spray nozzles. A set of stainless-steel eliminators are located above the spray-nozzle headers to eliminate moisture from being drawn into the fans.

A method of arranging the cold diffuser, accumulator, thermostatic control, liquid-control valve *C*, and refrigerating unit *E* is shown in Fig. 258.

Almost all foodstuffs while in cold storage lose weight owing to the low relative humidity of the air. Therefore, in order to prevent shrinkage, the relative humidity should be increased to about 85 per cent. Further increase in relative humidity, however, will likely cause spoilage. In order to carry a high relative humidity, it is necessary to have the temperature of the air leaving the spray chamber within 1° F. of the saturation point.

This is because a larger difference between the dew point and the allowable room temperature will produce little or no refrigerating effect.

The relatively narrow range in temperature difference between room and refrigerant for maintaining different relative humidities in the storage room is shown by the following table:

Relative Humidity, Per Cent	Temperature Difference, Room and Refrigerant, Degrees Fahrenheit
80	9.6
75	11.5
70	13.6
65	15.9

The coefficient of heat transfer between refrigerant and the brine spray is about 140 B.t.u. per square foot per hour per degree Fahrenheit.

Ventilation.—The same arrangements that will afford good air circulation in a cold-storage room are not sufficient for ventilation. Ventilation is a subject by itself and must be provided for separately, no matter what has been done in the way of securing circulation for maintaining even temperatures and in making it possible to rid the air of moisture. Ventilation must actually change the air, substituting fresh for foul.

Mold and many of the impurities of air found in cold-storage rooms, such as the decomposition gases given off by the produce, are not absorbed by the air but remain suspended in it. To get rid of these gases, the air must be changed by ventilation.

As so much in refrigeration depends on the conditions of the air as to temperature, humidity, and purity, cold-storage rooms cannot be ventilated simply by admitting a quantity of air from the outside, as in ventilating a house. The new air must be at about the same temperature and the same humidity as the air already in the cold-storage rooms, and it must be pure. A final consideration is that the amount of fresh air admitted to a room must be equal to the amount of foul air removed.

It is true that some storage plants are ventilated by the simple method of flooding the compartment with fresh air at intervals by opening the door. This arrangement, however, is slipshod, and the results are certain to be unsatisfactory. During the summer months, the outside air is much warmer than that inside the storage room, and it is likely, furthermore, to be carrying a large percentage of moisture. Such air, on being admitted to a

cold-storage room, gives up its heat very quickly, and some of the moisture it carries is condensed on the door and walls. On account of the rapid drop in temperature, the new air becomes saturated and raises the humidity of the compartment. By this method, then, heat is transferred to the goods, and the humidity is raised.

Ventilation is frequently left to take care of itself by leakage of air around doors and windows. Vent flues are constructed in the walls to conduct the foul air out. This method, too, is unsatisfactory. The rate at which air will leak into the room and at which the bad air will pass out depends on the difference in temperature between the air outside and the air inside. Thus, the efficiency of such a ventilating system depends on conditions of weather and season. When the outside air is warm, the vent flues will draw downward instead of upward, and to overcome this difficulty, it is necessary to heat the impure air before it will rise and discharge outside through the flues. Still another objection is that the incoming air is frequently humid. In leaking through the insulated walls, it loses temperature as it approaches the cold chamber, and in losing temperature, it condenses its water vapor which, by producing dampness, is injurious to insulating materials.

The wise cold-storage engineer, instead of making his ventilation depend on leaky walls, builds the cold-storage rooms as nearly airtight as possible. He then installs at the air intake a few simple pieces of ventilating apparatus—a fan, a cooling coil, a steam coil, and a few pans of calcium chloride. The cooling coil and the heating coil regulate the temperature of ingoing air, no matter what the weather or what the time of year.

The fresh air is admitted to the intake from a source as high above the ground as possible, and if it contains impurities, it is washed of dust by being passed through a spray. It is then cooled to a temperature from 8 to 10° F. lower than the temperature of the storage room. This increases its relative humidity so that it condenses some of its moisture on the cooling coils for the air supply. Then, on being passed over pans of calcium chloride, it gives up still more moisture. Now it is ready to be admitted to the circulation of the storage room, where its temperature will be raised, still further lessening its humidity. This process of ventilation should be gradual but continuous and should go on at such a rate that a week will be required to renew entirely the air of the storage room.

CHAPTER XIII.

COLD STORAGE OF FOODS

Cold Storage of Dairy Products.—In the storage of dairy products, temperatures ranging from -10 to 38° F. are generally used. Wholesalers who collect cream from farmers usually store it in refrigerators at a temperature of 36 to 38° F. until a large quantity has been collected. The cream thus kept is then shipped to large cities and made into butter which is stored at low temperatures (about $-10°$ F.) in rooms cooled by mechanical refrigeration. So stored, it will keep for a long time without spoiling. While it is possible to store butter at 36 to 38° F. for short periods of time, much better results are obtained by using lower temperatures. When butter is of good quality and properly packed, it can be kept 5 months without spoiling at temperatures of -12 to $-15°$ F.

Butter contains animal matter in the form of fats and may become rancid or moldy. To prevent its becoming rancid, it is necessary to exclude warm air and keep the temperatures low. If low temperatures are used, there is little need for excluding air; yet it is advisable to seal butter in packages as nearly air-tight as possible.

Mold which attacks butter is the result of storage in a warm and damp atmosphere. The remedy is to store butter at a low temperature and exclude air. When the air in the storage room is too dry, the butter loses weight as its water evaporates. It is, therefore, necessary that the air be more or less damp, and, for this reason, it should come into contact with the butter as little as possible.

Butter is generally stored in wooden tubs, made of either white spruce or ash. The wooden tubs are lined with paraffin and parchment paper before being packed with butter. When they are stored in a damp place, considerable moisture is absorbed, thus producing a growth of mold. Water promotes a growth of mold, while brine prevents it from forming. If the tubs have already acquired considerable moisture as a result of being stored

in a damp cellar or in improperly constructed cooling rooms, soaking the tubs in brine will be of little value in controlling mold. A better practice is to soak the parchment paper in the brine.

Butter.—Freshly made butter always contains some buttermilk. When the butter is being prepared the buttermilk should be removed. This is accomplished by washing and working the butter in water. Working it too thoroughly tends to spoil its grain and should, therefore, be avoided. When the buttermilk is fully removed, salt should be worked into the butter to take its place. A little extra salt should be added so as to take care of the loss during the working.

The latest methods used in packing are to pack butter which has been well worked in tubs which will hold 60 pounds. Tubs of this capacity are conical in shape, being a little larger at the top than at the bottom and are provided with covers made of the same wood. These tubs should be lined with parchment-paper liners cut to the proper size and shape. The butter is then packed solidly in the tubs to eliminate air holes. It has been found best to press it into the tub from the center outward toward the sides.

Creamery butter is generally sold in the form of "prints," each weighing 1 pound. In general, when butter is placed in cold storage, there is no certainty how long it will remain; it should not, therefore, be made in prints, for prints do not keep well through long periods of time. The working necessary to make prints breaks down the grain of the butter and injures its keeping quality. For these reasons, butter should be stored in bulk and should be made into prints only for short periods of storage or for immediate sale.

Ventilation of Butter-storage Rooms.—Butter stored in tubs oxidizes gradually, causing an accumulation of gases and odors in the storage rooms. To prevent this accumulation, it has been found necessary to ventilate compartments used for butter storage. Rooms containing butter packed in nearly air-tight packages need but little ventilation, while rooms containing tub butter need considerably more. In large butter rooms, the air is kept in circulation either by a fan located in a bunker for cooling the entering air, or the air may be cooled in the room by means of a cold diffuser which is a unit consisting of a fan and a refrigerating surface. Several cold diffusers may be located at con-

venient places in a large room so as to prevent air temperatures that are higher than should be used. The temperature of such a room should be about 0° F. Since butter will absorb odors it should be stored alone, and the humidity may be quite high for the temperature. The walls of the room should be covered with a suitable whitewash and at all times kept clean and free from dirt.

Cooling Milk and Cream.—Milk contains bacteria which increase rapidly unless kept at a temperature of about 40° F. Farmers who supply milk to creameries have usually no facilities for cooling milk, although the dairies of the milk companies offer the farmers better prices for milk when properly cooled at the farm.

Milk received in creameries is generally pasteurized. By *pasteurization* is meant the heating and holding of milk for 30 minutes at a temperature of 142 to 145° F. and then quickly cooling it to 38° F. This destroys the active bacteria. *Flash pasteurization* is the heating of the milk quickly to a temperature of 160° F. and continuing this temperature for a period of 8 seconds and then cooling to 38° F.

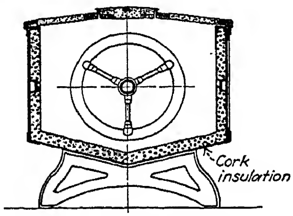


Fig. 259.—Helical coil type of holding pasteurizer.

Pasteurizers.—At present there are several types of pasteurizers on the market. These machines differ in the methods used in agitating the milk or cream while being heated or cooled. The horizontal-coil type of *holding*[1] *pasteurizers*, as shown in Fig. 259 consists of a rectangular or square tank made of tinned copper, nickel, chromium steel, or glass-lined steel. The sides, top, and bottom are insulated with a suitable insulating material with a steel or wooden casing outside the insulation. The tank has a cover made in a like manner. A helical coil is supported horizontally inside the tank. This coil is made of nickel or other suitable material and consists of hollow tubing having a diameter of 2 to 4 inches. This coil is supported on a shaft and directly connected to the shaft at both ends to permit the water to enter at one end and leave at the other. The shaft is driven by means of gears, thus causing the coil to rotate and agitate the milk or

[1] The *holding pasteurizer* differs from the *flash* type by the former retaining the milk for a longer time and at a lower temperature.

cream while it is being heated or cooled. A small tank is generally provided at the end of the vat to hold water that is used for heating or cooling.

Another type of pasteurizer is shown in Fig. 260. The only difference in principle is the manner in which the milk passes through the pasteurizer. In this holding pasteurizer the milk is continually but slowly moving through a series of tubes running from end to end but connected so as to form a continual path for the milk to pass through while being heated. This type, known as the "long-flow" type of pasteurizer, is quite commonly used in large milk plants.

A pasteurizing equipment for a dairy is shown in Fig. 261. The figure shows three sizes, namely, from left to right, 100-,

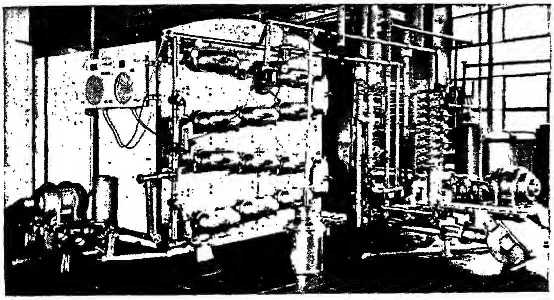


FIG. 260.—Long-flow type of holding pasteurizer.

200-, and 330-gallon tanks. These tanks are made low so as to permit easy and thorough cleaning. This type of pasteurizer may be used also for aging ice-cream mixture, for cooling and standardizing certified or raw milk, or for ripening cream. The 100-gallon tank is designed with a one-piece screw elevated cover and heating or cooling coil, thus making it suitable for creamery work. The 200-gallon tank is shown with a one-piece cover with spring-plunger counterbalance. The agitator and thermometers lift out of the tank with the cover. The 330-gallon unit is equipped with a two-piece cover.

These pasteurizer vats or tanks are made of seamless open-hearth steel and lined with acid-resisting blue-glass enamel. The walls are jacketed, the jacket extending below the liquid level, and the space outside the jacket is filled with 1 inch of cork board. The jacket may be used with steam for heating or with

brine for cooling. The contents of the vat are agitated by means of a three-blade propeller driven by an electric motor.

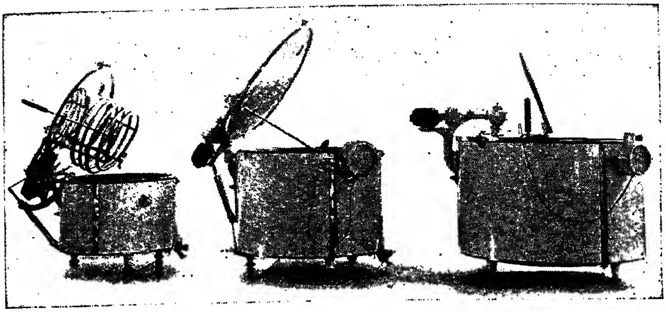


FIG. 261.—Dairy pasteurizing equipment.

A typical milk cooler is shown in Fig. 262. In this apparatus cool water may be admitted at C and discharged at D, while cold brine enters at A and leaves at B. In some milk plants, the pipe fittings at B and C are joined, and cold water is circulated in the pipes. This practice is resorted to during the winter months

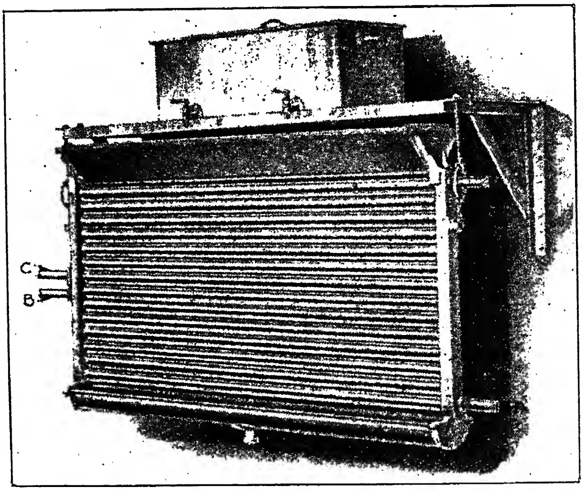


FIG. 262.—Typical milk and cream cooler.

when there is plenty of cold water available for cooling purposes. The milk to be cooled flows over the pipes from a trough located

at the top. In the bottom of the trough are located the openings through which the milk passes to the cooling surface.

When a centrifugal separator is used to separate the cream from the milk, the latter is taken from the pasteurizer after having been held at 145° F. for 30 minutes. The cream and sometimes the skimmed milk, after separation, are often cooled to 40° F. The cooler used to cool cream, because of the increase in viscosity, must be theoretically about 50 per cent larger than a cooler used for milk. In practice, however, a cooler for cream is generally made about 100 per cent larger than a cooler used for milk.

Milk may also be properly cooled by means of direct-expansion cooling coils, and such coolers are known as *direct-expansion milk coolers*.

The amount of heat which must be removed from milk is equal to the product of its weight, its specific heat, and the number of degrees through which it is cooled. The specific heat of milk is about 0.9 B.t.u. per pound per degree Fahrenheit difference in temperature. If, for example, 1,280 gallons of milk are delivered to a milk company every morning and this milk is cooled from 72 to 38° F. in 4 hours, the capacity of the refrigerating plant required to do this cooling is calculated as follows: Each gallon of milk weighs about 8.59 pounds; then $1{,}280 \times 8.59$ or 10,995 pounds of milk are to be cooled. The amount of heat to be removed is $10{,}995 \times 0.9\ (72 - 38)$ or 336,447 B.t.u. in 4 hours.

Since this heat is to be removed in 4 hours, the refrigerating system must be capable of removing per hour $336{,}447 \div 4$ or 84,112 B.t.u. The compressor must, therefore, have a refrigerating capacity per 24 hours of $84{,}112 \times 24 \div 288{,}000$ or 7.0 tons.

Now suppose that a compressor of smaller capacity than the above is installed and that its refrigeration is "stored" in relatively large brine-storage tanks. It will then be necessary for the smaller compressor to produce the same amount of refrigeration as the larger one in 10 instead of 4 hours. This smaller compressor would then have a capacity of $336{,}447 \div 10$ or 33,644 B.t.u. per hour, or a rated refrigerating capacity per 24 hours of $33{,}644 \times 24 \div 288{,}000$ or 2.8 tons.

Thus, a 3-ton plant with large brine tanks for the storage of refrigeration will do the same amount of cooling as a 7-ton plant. These calculations show the refrigerating capacity needed for the

actual cooling and do not allow for losses of the compressor. Because of this, the actual required capacity of the compressor should be about 25 per cent larger.

Suppose that the plant is running while the milk is being cooled; the plant then would remove from the milk during this time about 4 × 33,644 or 134,576 B.t.u. This leaves for the brine to remove 336,447 − 134,576 or 201,871 B.t.u.

If a calcium-chloride brine is used in the brine tank and the brine has its temperature raised 18° F. in the removal of heat from the milk, the weight of the brine needed for the brine tank can be found when its specific heat and specific gravity are known. In Table IX of the Properties of Calcium Chloride Brine (p. 253), the specific heat of brine having a strength of 22 per cent is 0.71, and the specific gravity is 1.2. Since 201,871 B.t.u. are removed by the brine, the weight of the latter is then 201,871 ÷ (18 × 0.71), which is 15,796 pounds, or, in gallons, it is 15,796 ÷ (8.33 × 1.2) or 1,580 gallons. Since it is important to know how large to build the brine tank, the volume of the brine in cubic feet may be found by dividing 15,796 by the weight of a cubic foot of brine which is equal to the product of 62.4 × 1.2 or 74.88 pounds; or the volume of the brine is 211 cubic feet.

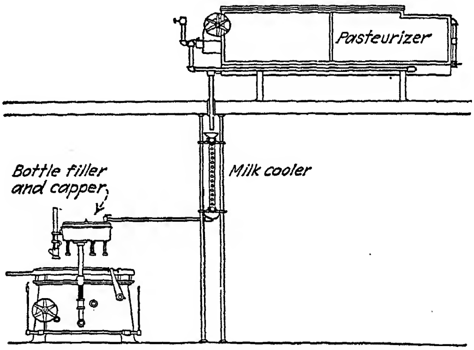


Fig. 263.—Simple layout of pasteurizer, cooler and bottle filler.

In the above example, the initial temperature of the milk is taken as 72° F. However, if the refrigeration is needed for milk just drawn from the cow a much higher temperature must be used. This temperature is about 90° F.

After the milk has been weighed, filtered by a filter generally located between the preheater and the "holding" section of the

pasteurizer, and then pasteurized and properly cooled, it is ready to be bottled. A simple manner in which to arrange the equipment is shown in Fig. 263. The milk after being cooled enters the bottling machine. The filled bottles are then put in cases and placed in refrigerated rooms held at about 38° F. These rooms may be cooled by unit coolers or cold diffusers, brine coils and direct-expansion coils. Frequently the cases are covered with cracked ice. This ice serves to keep the milk cool during delivery in the summer and acts as protection against freezing in the winter months.

A modern small milk-plant layout is shown in Fig. 264, while in Fig. 265 the construction and arrangement of the milk pasteurizing and bottling equipment may be seen. At the right,

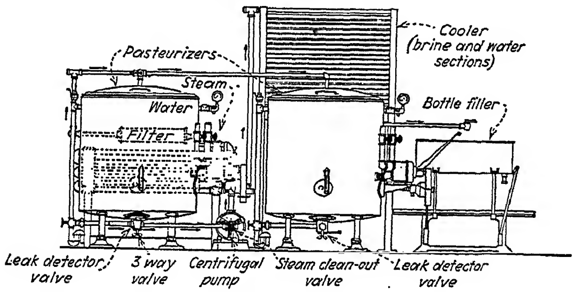


Fig. 264.—Equipment for a small modern milk plant.

against the wall, is the milk heater; in the center are two holding tanks *A* and *B* from which the milk is pumped to the milk cooler *C* suspended from the ceiling of the room. From this cooler the milk flows through a trough into the bottling machine *F*, where the bottles are filled and capped.

Transportation of Milk by Tank Car.—The modern dairy problems are many and quite complex, and one of these problems is the hauling of milk from great distances. This has been accomplished by the use of glass-lined tank cars as shown in Fig. 266. The construction of the car is along the lines of the standard-refrigerator type but has special insulation. Each car is equipped with two glass-lined tanks. These tanks are built in two sizes, namely, 3,000 and 4,000 gallons, thus making the total capacity of a car 6,000 or 8,000 gallons. The tanks are insulated with 2 inches of cork board which is enclosed in a sheet-metal casing. In transportation of milk over distances of

200 miles, the temperature of the milk varies less than 2° F. Each tank is provided also with a direct-motor-drive agitator. This type of car is used also in the transportation of condensed milk, ice-cream mixture, grape juice, and liquid sugar.

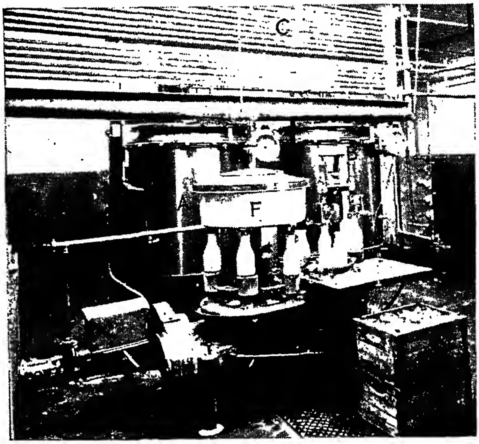


FIG. 265.—Typical milk pasteurizing and bottling equipment.

Freezing Ice Cream.—Whenever ice cream is made in bulk, mechanical refrigeration is used to cool and then to freeze the ice-cream mixture into ice cream, and finally to harden the ice cream. Generally, this method is cheaper and quicker, and the quality much better than the old one of freezing by means of a mixture of ice and salt. The latter method is now scarcely ever used. A modern ice-cream plant is shown in Fig. 267.

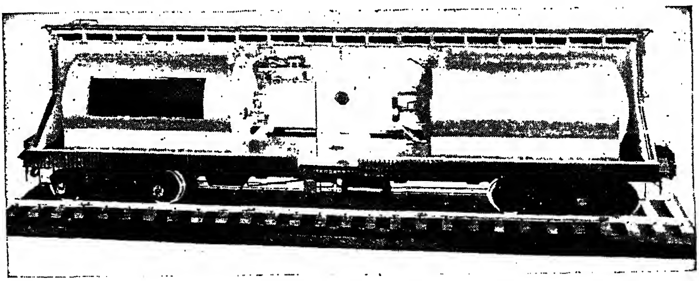


FIG. 266.—Insulated glass-lined tank car for transportation of milk, ice-cream mixture and other liquids.

The materials used in the manufacture of ice cream vary according to the adopted formula, but in all cases commercial ice cream must contain butter fat, or the fat of milk, the amount of which is generally between 7 and 12 per cent by weight. A so-called "binder" or "stabilizer" is added in the form of gelatin which tends to prevent the formation of ice crystals. In addition, the ice-cream mixture contains milk solids, extracts for flavoring, and sugar. In general, ice cream should contain at least 12 per cent (by weight) of sugar. The sugar content improves the smoothness of the ice cream. A very good ice cream can be made when 14 to 16 per cent (by weight) of sugar is used.

The ice-cream mixture is prepared in the following steps: (1) proportioning the ingredients that go into the mixture; (2)

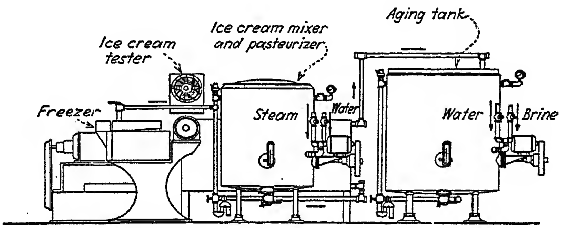
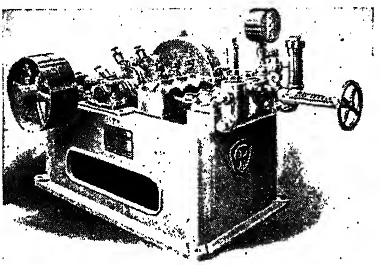


Fig. 267.—Typical ice-cream plant.

pasteurizing the mixture; (3) homogenizing the mixture; (4) aging the mixture. After having prepared the ice-cream mixture, it is pasteurized at a temperature of 145° F., the pasteurization continuing for 30 minutes. The mixture is then homogenized, after which it is cooled to a temperature of 40° F. It is aged for a period of 12 to 48 hours in insulated holding vats. Properly designed vats will hold the mixture for 18 hours without more than a 5° F. rise in temperature and without the aid of refrigeration. Aging causes the mixture to become smooth, glossy, viscous, and improves the smoothness of the finished product. However, the mixture should not have so much viscosity that the desired "swell"[1] or "overrun" cannot be obtained. In homogenizing, the viscosity is increased, but the increased viscosity due to aging does not produce the same result, as one is a mechanical action while the other is a chemical or colloidal action.

[1] The swell or overrun is the increase in volume which takes place during the whipping period and runs as high as 50 to 100 per cent.

Machines used to produce a homogeneous product such as milk, cream, and ice-cream mixture are known as *homogenizers* and the process is called homogenization. Homogenizers, in general, consist of a single-acting triplex pump, as shown in Fig. 268. Each of the three cylinders has a suction and a discharge valve. The discharge valve of each cylinder connects into the main discharge line in which is located a special valve. This special valve is constructed so as to permit the operator to vary the pressure required to force the fluid through it. The object of the homogenizer is to reduce the size of the fat globules. This process also increases the viscosity of the milk or ice-cream mixture. Milk that has passed through a homogenizer does not show the cream line as readily as milk that has not been subjected to this treatment. This is attributed to the fact that the fat globules do not rise as readily to the surface after having been broken down. It is thought by some authorities that homogenization influences the physical structure of the casein in the milk or cream. Milk for the trade is seldom if ever homogenized, as this process destroys the creaming ability of the milk. In the production of dry milk, however, the homogenizer is used when it is desired that the creaming properties of the reconstructed milk be destroyed. When producing ice cream the ice-cream mixture after being pasteurized is passed through the homogenizer, which produces a smoother texture in the finished ice cream than can ordinarily be obtained. Also the ice-cream mixture can be whipped in the freezer without the danger of churning the fat globules. The pressure generally used is about 2,500 pounds per square inch, and in some homogenizers which have two stage valves the pressure is reduced to about 1,500 pounds per square inch by the second valve. This procedure reduces the viscosity of the ice-cream mixture which occurs at higher pressures.



268.—Homogenizer for ice cream.

The freezing process is carried on in an ice-cream freezer, where heat is extracted from the mixture by either brine or ammonia cooling coils. This process is divided into two steps: (1) the freezing period and (2) the whipping period. During the freezing

period, the refrigerant should produce a temperature difference of about 18° F.; that is, the temperature of the mixture should be lowered from about 40° F. to 22° F. in not over 5 to 6 minutes, so as to produce a mixture of the proper consistency. The refrigerant is then shut off and the mixture of ice-cream mixture and frozen ice crystals is whipped for 3 or 4 minutes to give it the necessary swell or overrun. The whipping period should not be continued too long as the heat developed by the electric motor will cause the temperature of the mixture to rise and partly undo the freezing. Ice cream should be removed from the freezer at a temperature higher than 23° F. Having partially frozen the ice-cream mixture, it must then be hardened. The hardening is carried out in a hardening room, the temperature of which is held at a temperature of about −15 to −10° F. The ice cream should attain a temperature of 0° F. in a period of about 10 hours. The circulation of air in the hardening room is very important and care should be taken to stack the filled cans so as to allow the air to circulate between the cans. Recently, it has been considered good practice to use fans in hardening rooms, these fans to be so placed as to move the air upward toward the refrigerating coils. Ice cream that is hardened over a long period of time will not have the same excellent texture as that which is hardened rapidly.

There is a tendency to produce package ice cream which is frozen by conveying the packages through a chamber held at −30 to −60° F. The air in this chamber removes the heat so rapidly from the packaged ice cream that it hardens in a period ranging from 30 to 60 minutes. For such purposes direct-expansion coils are used, and the liquid refrigerant is forced to circulate in these coils by means of a suitable pump.

In the ordinary hardening rooms the latest practice is to provide sufficient refrigerating surface in the overhead and shelf coils to maintain the required room temperatures without reducing too much the suction pressure. Installations are made which have from 1 linear foot of 1¼-inch pipe for every cubic foot of room space, to 1½ linear feet of 2-inch pipe for each cubic foot of room space.[1] These figures vary with the size of the room. The larger the room the less the linear feet of pipe required. The

[1] Hardening rooms using the flooded system have usually one linear foot of 2-inch pipe per cubic foot of space. In some installations using the dry system there are 2 linear feet of 1¼-inch pipe per cubic foot of space.

present practice is to arrange the coils of 2-inch pipe on not less than 6-inch centers. Even greater spacing will aid the circulation of air.

Refrigeration Needed for Ice Cream.—The exact amount of refrigeration needed to produce ice cream is difficult to determine. This is because the specific heat of the ice-cream mixture before freezing, the latent heat of fusion, and the specific heat of the ice cream after it is frozen will vary with the quality of the ice cream.

The weight of frozen ice cream will depend on the increase in volume and the specific gravity of the mixture. If the swell or overrun is 80 per cent and the specific gravity of the mixture is 1.1, then the *weight per gallon* of ice cream, w, may be calculated thus:

$$w = 8.33 \times 1.10 \div (1.00 + 0.8) = 5.1 \text{ pounds}$$

In general, ice cream containing only extract flavors will weigh about 5 pounds per gallon, while ice cream containing fruit and nuts will weigh about 6 pounds per gallon. These figures depend on the amount of swell or overrun. If the specific heat of the ice cream before freezing is 0.78, the latent heat of fusion is 90, and the specific heat after it is frozen is 0.45, then the refrigeration per gallon can, for a freezing temperature of 23° F., be calculated as follows:

		B.t.u.
Cooling to freezing point	$= 1 \times 0.78(40 - 23) =$	13.26
Latent heat of fusion	$= 1 \times 90 \quad =$	90.00
Cooling after freezing	$= 1 \times 0.45(23 - 0) =$	10.35
Total refrigeration per pound	=	113.61

The refrigeration per gallon of ice cream containing only extract flavors is equal to 5.1×113.61 or 579.4 B.t.u., and for cream containing fruits and nuts the refrigeration is equal to 6×113.61 or 681.6 B.t.u. per gallon.

These figures are approximate as they depend on the type of freezer, temperature of cans, insulation, and other losses. If these losses are assumed to be equal to the refrigeration per gallon of ice cream, then the total refrigeration will be about 1,364 B.t.u. per gallon.

All of the refrigeration required to freeze the ice cream is not taken out in the freezer, and this amount may be considered as the heat required to cool the mixture to the freezing point plus

one-half of the latent heat of fusion. The remaining amount of heat to be removed to freeze the ice cream is removed in the hardening room.

Ice-cream Freezers.—There are two types of ice-cream freezers: (1) the type that uses brine and (2) the type that uses ammonia as the refrigerant. The latter is often called a "direct-expansion freezer."

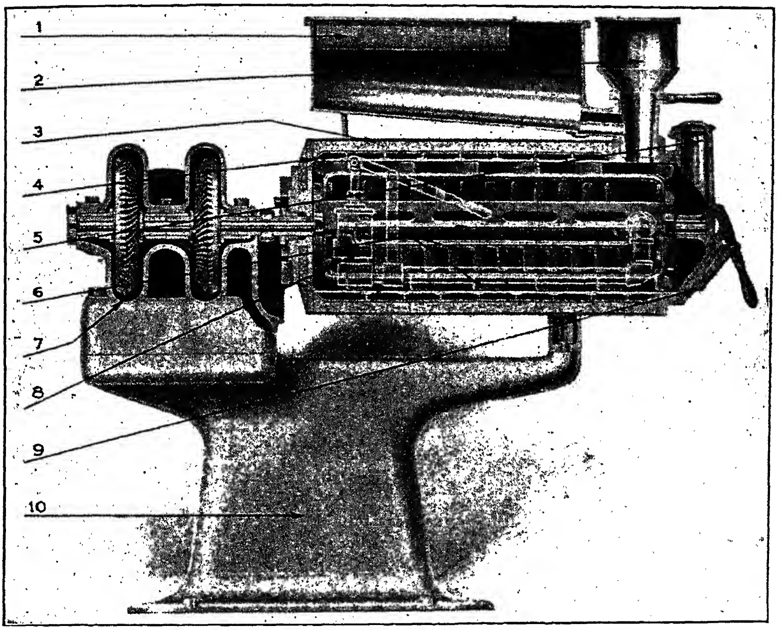


FIG. 269.—Section of brine-cooled ice-cream freezer: 1. strainer in mixing tank; 2. fruit funnel; 3. heat-insulating material; 4. spiral passage for brine around freezing cylinder; 5. peep hole; 6. blades of dasher; 7. driving gears; 8. "phantom" view of brine valve. 9. Opening for removing ice cream.

The ice-cream freezer shown in Fig. 269 uses brine and is made in sizes varying from 40 to 120 quarts. This freezer consists of a horizontal cylinder around which is a coil-like space through which the brine passes. Inside of the cylinder are dashers which are driven by bevel gears, the gears being driven by a motor or belt. The blades are held parallel to the axis and have contact with the cylinder walls for the entire length of the blade. They clear the wall of the mixture as fast as it

is frozen. The mixture enters through a strainer in the mixture tank, while the fruit is put into the freezer at the fruit funnel, thus insuring the proper freezing of fruit ice cream. A three-way brine valve located in the brine line controls the brine supply and also permits the maintenance of a constant brine pressure at the brine pump. The temperature of the brine should be about -15 to $0°$ F. The amount of brine circulated varies; but a good rule to follow is to have 4 gallons of brine per minute circulating through the freezers for each gallon of ice-cream mixture to be frozen. A peep hole is located at the front of the freezer which permits the operator to see the condition of the mixture while freezing. The partially frozen mixture is drawn off through an opening at the front of the freezer. It is advisable to allow about 3 tons of refrigeration for each 40-quart freezer.

The direct-expansion ice-cream freezer consists of a horizontal cylinder in which are placed the dashers. The cylinder has a jacket in which the ammonia evaporates. The dashers are driven by bevel gears which are driven by an electric motor. A vertical steel chamber or cylinder is mounted alongside the freezer as is shown in Fig. 270. This cylinder is an accumulator which is to prevent liquid ammonia from being drawn into the compressor as this freezer operates upon the flooded system. Inside of the accumulator is a float valve which regulates the amount of liquid ammonia entering the freezer, thus maintaining the proper liquid level. The liquid line is connected to the bottom of the accumulator, while the suction line is connected to the top of the accumulator. A nipple extends through the shell of the accumulator, so that vapor with entrained moisture will strike the opposite wall, causing the liquid to fall and leaving the vapor to pass upward through the hand lever shut-off valve and the automatic pressure-regulating valve in the top head of the accumulator.

Since the temperature varies with the pressure, the gage reads directly the temperature, which is the temperature of the ammonia in the freezer. The proper operating pressure will depend upon the character of the mixture being frozen and the method of whipping preferred. Usually, the pressure at the freezer is about 15 pounds per square gage. An adjustable back-pressure regulating valve located at the top of the accumulator maintains a constant ammonia pressure in the freezer. If the compressor suction pressure is about 12 pounds per square inch gage, satisfactory control and proper conditions for freezing are obtained.

When whipping, the lever should be pushed so as to close the suction line, and the pressure inside will rise to about 30 to 35 pounds per square inch gage. A safety-relief valve located in the top of the accumulator prevents an excessive pressure from developing if the hand valve should be closed for a long time.

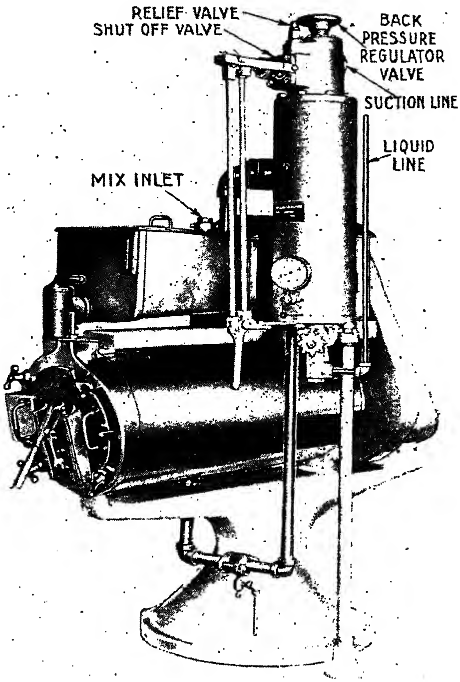


Fig. 270.—Direct-expansion ice-cream freezer.

Delivery of Ice Cream.—After the ice cream is hardened in the molds or cans it is kept in the storage room until needed for delivery. Today ice cream is usually delivered by means of trucks which are properly insulated and refrigerated by means of salt and ice, solid carbon dioxide or small mechanical refrigerating units driven either independently or from the engine used to drive the truck. In the case of house delivery, the ice cream is often shipped in corrugated paste-board cartons having several liners. The refrigeration for these boxes is provided

by means of slabs of solid carbon dioxide, cut about 1 inch

in selling small quantities of bulk or brick ice cream, cabinets, as shown in Fig. 271, are commonly used. These cabinets are built with two or more holes for the ice-cream cans. They are equipped with a suitable refrigerating unit which is automatic and generally air-cooled. The cabinet shown is insulated with 3 inches of cork board on the top, 3½ inches of cork board on the sides, and 4 inches of cork board on the bottom. Direct-expansion coils are shown embedded in a special cement known as "Copeman stone." This material is molded to size so as to hold the ice-cream cans without a large air space between the

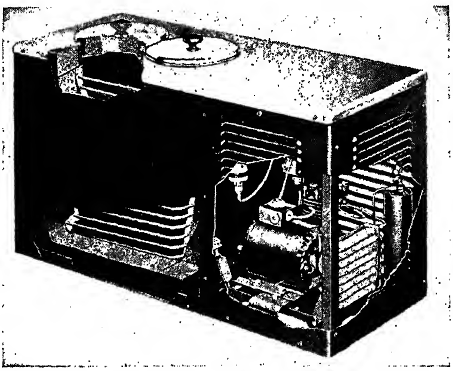


FIG. 271.—Refrigerated cabinet for ice cream.

cans and the Copeman stone. The cabinet is held at a temperature of about 0 to 6° F.

Storage of Cheese.—At present the processes used in the manufacture of cheese are undergoing many changes; these changes are based upon the application of fundamental facts known about the chemistry and the bacteriology of milk. Cheese contains the constituents of milk in a concentrated form which are chiefly, fat, casein, and insoluble salts, water which contains soluble salts, lactose, and albumin. In order to bring these constituents into concentrated form, the milk is coagulated by lactic acid produced by bacteria or by adding rennet. Rennet or rennin is the substance that is commonly used in the home to make what is known as "junket." The junket tablets consist of powdered rennin. After the curd is formed, the water is removed

The cheese may or may not be ripened, depending upon the process and the kind of cheese to be made.

Many types of bacteria are present in the curd and these are influenced by the temperature and humidity at which the cheese is held. The control of temperature and humidity are, therefore, important features in the ripening process of cheese. It is the opinion of some cheese makers that cheese should be ripened at 60 to 70° F., but extensive tests show that the flavor is improved by the ripening of cheese at lower temperatures. The rooms should be arranged for refrigeration control as some cheese may be mold cheese and some not. Some cheese will require a high relative humidity and others less. It is important that the air in one room will not mix with the air in another, as the molds would then become mixed. In order to meet these requirements air conditioning should be resorted to, and the temperature of the washed air should be about 40 to 45° F. The air leaving the washer is saturated but when mixing with the warmer air in the curing room the relative humidity drops. Therefore, in some cases, it is necessary to control the relative humidity, which may be done by adding moisture. This may be accomplished by the use of steam jets in the coolers.

Generally, cheese that is held at too high a temperature during ripening develops too much acidity which causes a sour-flavored product. On the other hand, if the humidity is too high, molds are likely to develop which will cause the cheese to have a rancid taste. If the room humidity is too low the cheese will lose weight, and to prevent this loss in weight it is often coated with paraffin.

Roquefort when held at too high a temperature becomes overrun with air-borne contamination of yeast and molds; this causes the curd to break down producing a salvy and oily texture. With the temperature controlled, it is possible to stop the Roquefort mold at the right point to obtain the proper flavor and aroma. This kind of cheese should be held at 45 to 50° F. and a relative humidity of 83 to 90 per cent during the early stages of ripening. Later, the temperature should be lowered to 40° F. with a relatively lower humidity. When the ripening process has been completed, the temperature for holding should be just above freezing.

In the case of Swiss cheese the temperature for ripening is approximately 70° F. for the early stages, and, later on, the

temperature should be lowered to about 40° F., which is necessary to produce the characteristics of this kind of cheese.

Cheese is generally stored in boxes which contain about 60 pounds, and these boxes occupy about 2 cubic feet.

Cold Storage of Meat.—At present the refrigeration of meat is going through a period of alteration as the progress made in quick freezing has introduced new and somewhat revolutionary methods of chilling, freezing, and holding of meats.

The application of refrigeration to the chilling of meat calls forth various methods depending upon the particular purpose required. The most important step in the refrigerating of meat is the chilling of meat after it has left the killing room. The blood temperature of fresh-killed animals such as cattle and swine are, respectively, about 100½ and 103° F. As soon as the carcass is prepared, it should be placed in the chilling room and its temperature lowered. The temperature of this room is about 26° F. This will prevent deterioration of the flesh by bacteria, as the temperature of the carcass is close to the optimum for their growth. The above cooler temperature will bring the internal temperature to about 34 to 36° F. The time required for this process is about 18 to 24 hours.

During the chilling period a rapid circulation of air about the carcass must be maintained. This air may be chilled by a loft above the chilling room in which are located adequate direct-expansion coils, the air circulating naturally, or the air may be cooled by means of a fine, cold brine spray through which the air passes, as is shown in Fig. 272. The chill room and lofts must be so designed as to have the proper circulation of air without any dead air spaces, otherwise the meat is likely to "sour."

There is shown in Fig. 272 a brine header in which there are spray nozzles through which the brine is forced at a pressure of 8 to 20 pounds per square inch gage. This pressure varies with the width of the loft. The nozzles may be placed 1 to 5 feet apart, on centers depending upon the amount of brine to be sprayed. The brine falls upon a floor which is made tight by roofing material and a layer of asphalt. The efficiency of this system depends to some extent upon the fineness of the spray. This system suffers from brine dilution because the moisture from the fresh-killed meat is absorbed in the air and then taken out of the air by the cold brine, and therefore care must be taken to maintain a constant concentration.

The temperature control of the chill room is quite important, and the ideal control would produce a prompt and uniform reduction in temperature. The differences in internal temperature of the meat in different parts of the room may be as great as 4° F. Because of this, the design and operating conditions should be carefully studied.

Meat stored for short periods does not require very low temperatures. A temperature level of 40° F. gives excellent results when the storage period is not above 2 or 3 months. If it is necessary to keep meat longer, it is best to freeze it, bringing its temperature down to 10° F. After this freezing, it can be stored at about 30° F. for periods up to 6 months and sometimes longer.

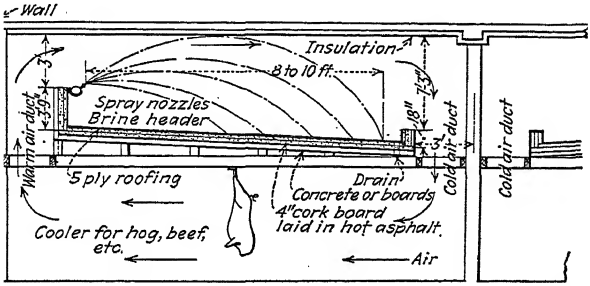


FIG. 272.—Chilling room with overhead loft cooled by spraying brine.

The recent developments in the field of quick freezing show that the quality of frozen meat after thawing depends upon the rate of freezing. If the heat is extracted rapidly, the cellular structure of the flesh is not broken down by the formation of the ice crystals as will result from the older and slower methods of freezing meat. Meat which is frozen slowly will be found to be pulpy and lacking in texture.

Possibly the strongest argument for rapidly cooling beef immediately after killing is that it reduces the possibility of "bone taint." This term is used to indicate certain putrefactive changes that occur in the neighborhood of the hip joints of the carcass which is characterized by a very unpleasant "gassy" smell.

When preparing the cuts for curing, the carcass should not be allowed to get too warm. The temperature of the room should not be over 45° F., and the cuts should move right along. If the

carcass is too cold, the process of cutting will slow up, and if the temperature is too high, the carcass is likely to start to spoil.

The cuts from the carcass are placed in brine at a temperature of 38° F., and the room is held at the same temperature by good air circulation. Direct-expansion coils may be used for this purpose. Wide variations of temperature greater than 2° F. should be avoided. Reducing the curing temperature retards the curing of the meat.

The amount of refrigeration varies and for chill rooms 80 B.t.u. of refrigeration per cubic foot of room space per 24 hours is required. In curing, about 40 B.t.u. is required per cubic foot per 24 hours.

To calculate the proper area of cooling surfaces by the amount of heat to be removed, it is necessary to understand all the conditions of temperature, insulation, etc. Siebel has prepared a few practical rules for pipe allowance, which will apply in most cases. The rules are given in the following table.

PIPING FOR COLD STORAGE OF MEAT

Type of room	Quantity and size of pipe allowed	Volume of cold-storage rooms, cubic feet of storage space	
		For direct expansion	For brine circulation
For chilling rooms	One running foot of 2-inch pipe (or its equivalent)	13 to 14	7 to 8
For storage rooms	One running foot of 2-inch pipe	45 to 50	15 to 18
For freezing rooms	One running foot of 2-inch pipe	5 to 10	3

Storage of Eggs.—The air in a shell-egg storage room should not be permitted to become too damp and warm, as a fungus growth rapidly develops on the eggs. It will cause them to become musty. Although dampness is largely responsible for this growth, its development can be checked by the maintenance of low temperatures. A relative humidity of 80 to 85

per cent is not likely to injure shell eggs if the temperature is kept at 29 to 31° F. Air that is too dry will absorb moisture from the whites as the shell is porous. An egg contains 70 per cent water and 30 per cent solid. Eggs in storage for a period of 6 months lose about 5 to 7 per cent of their total weight. A storage which has only natural circulation loses by evaporation of moisture about 3 ounces per case of 30 dozen eggs per month. The freezing point of eggs is about 28° F., and, if frozen, the egg shells are likely to crack. Frozen eggs in the shell are unfit for consumption.

Space Requirements for Foodstuffs
Meat rails placed approximately 30 inches on centers

Material	Average weight, pounds	Floor space, sq. ft.	Space occupied, cu. ft.	Clear height of room, feet
1 Barrel apples or potatoes.	180	4		
1 Tub butter.............	60	2.5	2.5	
1 Cheese..................	60	2	2	
1 Case eggs (30 dozen).....	70	...	3	
1 Beef....................	700	9	108	12
1 Sheep...................	75	2	16	8
1 Hog.....................	250			
1 Calf....................	90			

When shell eggs are kept at too high a temperature they will spoil as they are more subject to molds, bacteria, and enzymes, than shell eggs kept at the proper refrigerating temperature.

The proper ventilation and circulation of the air in the storage room are important because eggs during storage undergo a gradual decomposition, which results in the giving off of gaseous impurities into the air. The air in coming in contact with the surface of the refrigerating coils gives up some of its moisture by condensing, purifying the air of germs and decomposition products. Not all of the gases are disposed of in this manner and, therefore, a change of air is necessary. Where the circulation and control of humidity are good, new air needs to be added only at such a rate that a complete change of air will be required once in a week or every week and a half.

The circulation of an egg-storage room should be strong and rapid, for the eggs are not directly exposed to the air currents; they are packed in individual pockets in crates, and the crates

or containers are packed together and piled. Effective circulation, then, must reach each individual egg; and to do this, it must be strong and capable of being directed. Rapidity of circulation is desirable, because it makes possible adequate refrigeration with a small area of cooling surface and keeps the air dry at lower temperatures. Satisfactory results are difficult to obtain by any method of natural circulation. Forced circulation, as shown in Fig. 273 with its strong uniform air currents and its easy control of humidity, is favored in modern practice.

Shell eggs upon arriving at an egg-breaking plant are taken to a chilling room where they are kept at a temperature of 31° F.

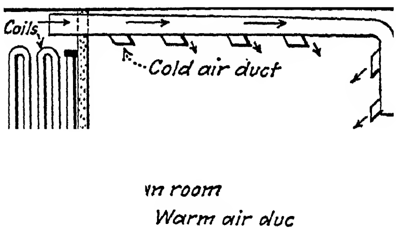


Fig. 273.—Forced air circulation for egg-storage room.

They are held at this temperature for 12 to 28 hours; this stiffens the whites and facilitates the grading and separating process later on. The broken eggs are put into 30-pound cans and frozen solid in 48 to 60 hours at a temperature of −5 to −12° F. and then kept at 0 to −5° F. Canned eggs are used extensively in the baking industry.

In the case of shell eggs ⅓ ton of refrigeration is required to cool 33 cases containing 30 dozen eggs from an initial temperature of 70 to 29° F., while for canned eggs 1 ton of refrigeration will cool 1 ton of canned eggs from an initial temperature of 70 to −10° F. in 24 hours. The temperature of eggs at the time they are placed in cold storage generally ranges from 70 to 90° F.

CHAPTER XIV

QUICK FREEZING

Quick-freezing Methods.—During recent years new methods of refrigeration have been introduced for preserving perishable food products, so that after thawing they will have their original properties of taste and appearance. A few years ago, the only food products to which the freezing methods to be described were applied were various kinds of fish. By the so called method of *quick-freezing*, meaning very rapid freezing at extremely low temperatures (-40 to $-50°$ F.) it is possible to preserve fish and some other perishable food products so that, when thawed and prepared for eating, they will have the flavor and appearance of similar fresh foods. Recently, such methods of quick-freezing have also been successfully used for refrigerating meats, poultry, vegetables, fruits, and fruit juices.

The relatively older method of slower ("sharp") freezing of perishable food products at temperatures of 0 to $-20°$ F. *in still air* produces an inferior quality; but, if these same food products are refrigerated by the improved methods of quick-freezing, it will be found, even after months of storage, that they will have, in most cases, the same qualities of flavor and appearance as they had before being refrigerated.

What Happens When Flesh Is Frozen.—All animal matter, whether fish, meat, or poultry, is composed of minute elastic-walled cells which are filled with a jelly-like fluid containing various chemical salts, especially sodium and calcium in solution. When the moisture in this jelly-like cell material with its salt content is refrigerated, it forms water-ice crystals which are of different sizes, the crystals beginning to take shape just as soon as the refrigerating temperature is lower than about 32° F. When the temperature is lowered still more, the formation of the ice crystals becomes increasingly rapid, leaving behind a more and more concentrated solution of the various salts, the latter freezing completely in some cases only when the temperature is reduced to about $-65°$ F. On the other hand, about three-

quarters of the moisture in animal matter is frozen into ice crystals before the temperature of the food is less than 25° F.

It is well known that the more slowly crystals are formed, the larger they will become. The size of ice crystals, for this reason, increases with the time required for freezing; meaning, also, that when foods are frozen very rapidly, the ice crystals will be relatively small. Since, however, nearly all the moisture of animal flesh is frozen in any method of refrigeration between the temperatures of 32 and 20° above 0° F., it is obvious that every effort should be made to pass such food products through this range of temperature as rapidly as possible, if the ice crystals are to be as small as possible. When food products are frozen

Fig. 274.—Cells of haddock frozen in usual way.

Fig. 275.—Cells of fresh unfrozen haddock.

Fig. 276.—Cells of quick-frozen haddock.

to low temperatures by other than the quick-freezing methods, a great many of the crystals may grow to many times the size of an individual cell; and these large crystals will break up the tissues, piercing and tearing the delicate cells very much, as shown in Fig. 274. The cells in this figure are to be compared with the structure of the cells of the fresh food, as shown in Fig. 275, and also with Fig. 276 showing the perfect condition of the cells when the same food is refrigerated by the recently improved methods of quick-freezing. Because of such breaking up of the tissues, as illustrated in Fig. 274, the methods of slow freezing make the food products so refrigerated unattractive in appearance and more or less lacking in flavor and food value. Slowly frozen food products also lose considerable weight and shrink in size during such freezing. They become discolored because the blood becomes oxidized. "Freezer-burn" (Fig. 277) is the

kind of discoloration due in the first place to a loss of moisture and, secondly, to oxidation. During the process of quick-freezing, there is practically no shrinkage, and the flavor, appearance, and keeping qualities are not essentially different from those of the fresh food, as shown in Fig. 278. In order to secure extremely rapid freezing, it is necessary to bring the product to be refrigerated into contact with (1) rapidly circulating cold air; (2) cold liquids or vapors; (3) exceedingly cold metal surfaces. The last named method is preferable as the flesh is then placed between two metal surfaces so that heat may be extracted simultaneously and in equal amounts from both sides. Similar results cannot be obtained by any method of freezing *in still air* for the reason that air which is not in rapid movement is a very poor

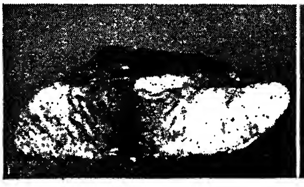


FIG. 277.—Sharp-frozen haddock showing "freezer-burn."

FIG. 278.—Quick-frozen haddock.

conductor of heat; and the refrigerating process in still air is, therefore, even with very low temperatures, relatively slow.

Quick-freezing as Applied to Vegetables.—Plant tissues are in many respects quite different from those of animals and behave differently in the freezing process. The cell walls of animal tissues are somewhat elastic as already explained and will not burst merely by the *expansion* of the contents of the cells during freezing. If the cells of animal tissues are ruptured during refrigeration it is because of the formation of large ice crystals that grow during any process of slow freezing. The cells of plant tissues, on the other hand, have comparatively inelastic walls which are very easily ruptured merely by the *expansion* of the contents of the cells during any kind of refrigeration, regardless of the rapidity of the freezing. Most vegetable tissues contain also a very large amount of intercellular moisture, and the expansion of this water during freezing has also a tendency to rupture the vegetable tissues. For the reason that vegetable matter cannot, therefore, be frozen by any method of refrigera-

tion so that the tissues are not ruptured, refrigerated vegetables cannot be made to resemble in flavor and appearance the fresh vegetables. It is claimed, however, that the cell rupture and other effects of freezing are in most cases an actual benefit to vegetable products which are to be cooked before they are eaten. One function of cooking vegetables is to break up the cell structure; and since any freezing process accomplishes this cell rupture to some extent, it may be expected that frozen peas, beans, asparagus, spinach, and similar vegetables will have to be cooked only about half as long as a fresh vegetable.

Quick-freezing of Vegetables.—The quick-freezing process when applied to vegetable matter does not offer the same advantages that are obtained when it is used to freeze animal flesh; but, nevertheless, there are advantages in the quick-freezing of vegetables over the method of slow freezing. The latter method results in the formation of very large ice crystals in the vegetable tissues and, therefore, causes considerably greater tissue damage and consequently more change of appearance and flavor than when the freezing is done very rapidly. With many of the more delicate fruits and vegetables any method of slow freezing produces a very distinct and appreciable change in flavor. In extremely rapid freezing of vegetable food products, on the other hand, the tissue damage is reduced to a minimum, and appearance and flavor are much better preserved than by any slow-freezing processes. There is also less tendency for vegetables to accumulate molds and similar organic growths on their surfaces.

Storage of Quickly Frozen Products.—Animal and vegetable products that are quickly frozen must usually be shipped long distances. Successful storing of such products and the efficient distribution of them are, therefore, important. Prolonged cold storage of either animal or plant tissues may cause damage by (1) excessive loss of moisture; (2) oxidation; (3) fermentation; (4) contamination from other foods and substances; (5) loss of volatile flavoring matter.

The excessive loss of moisture in food products is one of the most serious problems in refrigeration and is due to the escape of moisture vapor from the product through the air to the refrigerating medium. Such loss of moisture causes serious shrinkage, ruining the appearance and the flavor of the product that has been refrigerated.

Oxidation is another serious form of deterioration during cold storage and takes place rapidly when the product is freely exposed to the air in the cold-storage room. The higher the storage temperature, the greater, of course, will be the amount of oxygen in contact with the refrigerated product, and, consequently, the more rapid the oxidation will be. Oxidation may cause any one or all of the following bad effects: (1) rancidity; (2) color changes; (3) development of off-flavors. Fatty or oily products, such as salmon and pork, have a decided tendency to become rancid in cold storage. Fillets of fish, especially if very slightly "brined," are likely to develop a "salt-fishy" odor and taste. Many vegetables, such as spinach and asparagus, acquire a strong hay-like flavor and smell. Orange juice is similarly subject to changes in flavor.

All of the deteriorating changes mentioned take place more slowly as the cold-storage temperature is reduced; and for prolonged storage most products may not be safely stored if the temperature is higher than from 0 to 5° F.; and there is now a decided tendency toward still lower cold-storage temperatures, as for example, from −10 to −15° F. to make provision for the quick-frozen food products that are now being distributed.

Transportation Problems.—Quick-freezing of food products cannot be considered an entirely successful process until provision is made for the successful storage and transportation, without damage, of such products to the place where they are to be used. It is essential, of course, that thawing be prevented during transportation and until they are to be used. If quick-frozen food products are allowed to thaw for even a very short time and are subsequently refrozen, the quality of the food depreciates considerably. The formation of mold and other organic growth must be prevented as much as possible during transportation. With animal tissues this is not difficult as long as this food is kept "frozen hard," usually at about 20° F. On the other hand, both of these forms of damage act so rapidly on plant matter, even when still largely frozen, that the car temperatures for the transportation of vegetables and fruits which are to be in transit for a week or longer should not be higher than 10° F.

The only entirely safe way to distribute quick-frozen perishable foods is to make sure that they remain hard frozen until they reach the consumer. Such foods contain within themselves a large amount of refrigeration and if placed in well-insulated

shipping containers may be transported for long distances even by ordinary express or parcel post. For this purpose, corrugated fiber-board containers (p. 369) may be used. Such a container, having walls about 1 inch thick, will withstand shipment by express; and if solidly filled, will, even in hot weather, keep its contents completely frozen for from 3 to 5 days—and perfectly fresh, although partially thawed, for a somewhat longer period.

There are two fundamental facts of great importance in connection with the efficient transportation of frozen perishable foods. The first is that heat leakage into a container—whether it is an individual package or a refrigerated car—is in direct proportion to the surface area of the container, while the ability of the contents of the container to absorb that heat without thawing is in proportion to the weight of the food in the container. A solidly filled container, therefore, will absorb more heat with less thawing than a similar package less compactly filled. A compactly filled container that will keep a food product frozen for, let us say, 6 days at an outside temperature of 65° F. will keep the product frozen only 3 days when loosely filled with only half as many pounds of food.

Cost of Quick-frozen Packaged Foods.—To the casual observer it will seem inevitable that the packaging and quick-freezing of foods will increase the cost to the consumer. There are savings, however, in growing costs, packing, shipment, spoilage, and preparation in the home that more than compensate for the cost of packaging and freezing. Consider, for instance, the case of a meal of spinach consumed in a northern state in January. Under the present marketing conditions, approximately one-third of the volume of the spinach is inedible stems, discolored leaves, and foreign matter; and it must be loosely packed in baskets or crates and so placed in freight cars that air will circulate freely around all the packages. Thus only a comparatively small amount of edible product can be put into the car. After arrival in the retail store, the spinach spoils very rapidly, and in some stores this spoilage amounts to from 15 to 35 per cent.

CHAPTER XV

PRODUCTION OF SOLID CARBON DIOXIDE

For years the formation of solid carbon dioxide known commercially as "dry ice" was a laboratory experiment. For producing solid carbon dioxide in the laboratory a cylinder containing liquid carbon dioxide under pressure was used. To a valve on this cylinder a cloth bag was attached, and liquid carbon dioxide was allowed to escape into it. The rapid evaporation of the liquid at atmospheric pressure caused the temperature of the liquid to drop below the *triple point*[1] which is $-70°$ F. and thus form carbon-dioxide snow.

The increase in demand for solid carbon dioxide has been rapid. In 1925 the total amount used was about 300,000 pounds, while in 1928 this amount increased to 14,000,000 pounds. The year 1929 showed an increase of over 300 per cent for the year 1928. Further increases are reported for more recent years.

Sources of Carbon Dioxide.—Carbon dioxide or carbonic acid is found everywhere. It was first discovered by Priestley about 1770 by watching the process of fermentation in a brewery. Calcium-carbonate mountains contain it up to 45 per cent and over. Volcanic formations, springs, caves, and abysses of the earth give forth an abundance which is practically pure. All processes of fermentation which reduce the sugar in the original substance form practically equal parts of alcohol and carbon dioxide. In the combustion of coal, coke, oil, wood, and natural gas, large quantities of carbon dioxide are formed. The atmosphere contains between 0.003 and 0.004 per cent by volume of carbon dioxide.

Many industries, such as the alcohol and the chemical, develop pure carbon dioxide in a free state as a by-product. In alcohol plants the carbon dioxide is exceptionally pure but has certain odors which must be removed before it is suitable for use in manufacturing solid carbon dioxide. This purification process is simple, consisting of scrubbers and deodorizers in which a special

[1] The *triple point* of a substance is the temperature at which a substance be in the gaseous, liquid, or the solid state.

charcoal is used, the charcoal being reactivated at intervals by the application of steam.

One of the simplest methods of producing carbon dioxide consists in the calcination of limestone, such as magnesite, in closed retorts. At a temperature of 600° F., this substance loses its carbon dioxide, and after scrubbing by cool water is suitable for production. It is also possible to treat ordinary limestone in this manner, but a much higher temperature is required.

The chimneys of power plants discharge into the atmosphere large quantities of carbon dioxide. This gas contains large amounts of impurities which are contained in the ordinary steaming coals. To eliminate these impurities a grade of fuel containing a large percentage of carbon must be used. In the combustion of a fuel containing chiefly carbon, the carbon combines with the oxygen, forming carbon dioxide. One pound of carbon when combined with oxygen will produce about 3.67 pounds of carbon dioxide. This value in practice is never attained, and only about one-third of this amount is actually available. A high grade of coke such as "seventy-two hour" foundry coke, which may contain as high as 97 per cent of carbon and very little sulphur, is satisfactory. One pound of carbon requires $2\frac{2}{3}$ pounds of oxygen to form $3\frac{2}{3}$ pounds of carbon dioxide. Since air by weight contains 23 per cent oxygen, the amount of air required to supply $2\frac{2}{3}$ pounds of oxygen is equal to 11.6 pounds. At a temperature of 75° F., the volume of dry air required per pound of carbon is about 11.6×13.5 or 157 cubic feet. As air contains also 77 per cent nitrogen by weight, the amount of nitrogen supplied to the furnace per pound of carbon is equal to 0.77×11.6 or 8.93 pounds. As an excess of air is necessary in practice, the volume of air supplied is larger than the above value. This reduces the percentage by volume of carbon dioxide from 23 to 17 or 18 per cent, there being about 1 or 2 per cent of oxygen, the remainder, about 4 per cent, being inert gases.

The gas resulting from combustion, after having given up part of its heat in an exchanger or economizer, is scrubbed by water which cools and purifies it. The cooled and purified gas is then brought in contact with potassium carbonate which readily absorbs the carbon dioxide. The towers in which this process takes place are known as *absorber towers*. This part

of the system is quite inefficient as only about 50 per cent of the carbon dioxide in the gas is actually absorbed, the remainder being discharged to the atmosphere with the inert gases. If this process could be improved so as to reduce the percentage of carbon dioxide, absorbed from the gas, from 18 to 5 per cent instead of about 9 per cent, the efficiency would be materially increased and about 1.85 pounds of carbon dioxide would be formed per pound of coke.

Solidification and Sublimation of Carbon Dioxide.—The production of solid carbon dioxide involves first the liquefaction of the purified and cooled carbon-dioxide gas. This requires the gas to be compressed in several stages, cooled in several stages, mixed with low-temperature gas from the liquid coolers in stages, and finally its solidification. Owing to the large amount of heat added during the compression, it is necessary for economical reasons to carefully arrange the compression equipment.

When liquid carbon dioxide is expanded at atmospheric pressure, some of the liquid carbon dioxide changes into the solid state. The conversion factor for the solid carbon dioxide produced may be calculated from the pressure-heat diagram facing page 514 of the Appendix, as it is the ratio of the *latent heat of sublimation* (p. 440) to the available heat in the liquid. When the solid carbon dioxide changes from the solid to the gaseous state, it does so without returning to the liquid state. Carbon dioxide at atmospheric pressure cannot, therefore, exist in the liquid state. However, at the *triple point* (p. 437), it may exist in the gaseous, liquid, and solid states. The vapor pressure for the triple point is 75.1 pounds per square inch absolute, and the corresponding temperature is $-69.88°$ F. Because of this, carbon dioxide ceases to be a liquid below a pressure of 75.1 pounds per square inch absolute.

When the carbon dioxide in the solid state has the pressure lowered to that of the atmosphere, the temperature drops from -69.88 to $-109.2°$ F. Now, during sublimation, heat is being added, which is known as the latent heat of sublimation. This factor has been variously stated and according to Behn is 256.3 B.t.u. per pound, while Andrews gives 253.8 B.t.u., and Maass and Barnes 246.4 B.t.u. per pound. More recently, Plank and Kuprianoff mathematically determined the latent heat of sublimation to be 246.41 B.t.u. per pound. The *latent heat of sublimation* is made up of the total heat of the vapor above

fusion, the latent heat of fusion, and the heat of the solid. The latent heat of fusion when the liquid passes into the solid state has been found by Kuenen and Robson to be 78.8 B.t.u. per pound, and given by Plank and Kuprianoff to be 85.17 B.t.u. per pound. From the pressure-heat chart (p. 514) the heat of the solid is 14.27 B.t.u. per pound. The total heat above fusion found from the chart is equal to 146.97 B.t.u. per pound. Thus the latent heat of sublimation is equal to the sum of these heat values: $146.97 + 85.17 + 14.27 = 246.41$ B.t.u. per pound. It should be noted that these figures are not for the normal atmospheric pressure but for a pressure of 14.22 pounds per square inch absolute. The specific heat of the solid carbon dioxide is about 0.38.

If the temperature of the carbon-dioxide vapor is allowed to rise to 0° F., thus increasing the temperature of the vapor at atmospheric pressure 109° F., then there has been added the refrigerating effect of the heat of superheat. If the specific heat of superheated carbon-dioxide vapor at atmospheric pressure is taken to be about 0.19 B.t.u. per pound per degree Fahrenheit, then the added refrigerating effect is 0.19×109 or about 20.8 B.t.u., which should now be added to 246.41 and gives a total refrigerating effect of 267.21 B.t.u. per pound.

It has been shown that without the proper exchange of heat and pressure control in a solid carbon-dioxide plant the power required to produce 1 ton of solid carbon dioxide may be as great as 400 horsepower, while, if heat exchanges and pressure preservation is used, this power may be reduced about half.

One great advantage obtained when using solid carbon dioxide as a refrigerant is its ability when in the gaseous state to serve as an insulator. Tests have shown that when a block of solid carbon dioxide is surrounded by the evolved gases, the rate of sublimation (p. 439) was reduced about 17 per cent.

Solid carbon dioxide in its commercial form varies in color from white to a grayish translucent shade, becoming more translucent with increasing density. The density averages between 80 to 95 pounds per cubic foot in commercial 10-inch cubic blocks.

Cycles for Producing Solid Carbon Dioxide.—There are three distinct thermal cycles in the manufacture of solid carbon dioxide. These are the non-regenerative, the regenerative, and the complete re-expansion cycles.

The *non-regenerative cycle* consists of a first stage, a second stage, and a third stage of compression. Located between the first and second, and between the second and third stages are intercoolers used in order to reduce the horsepower required for compressing the refrigerant. The third-stage compressor discharges into the condenser where the refrigerant is liquefied.

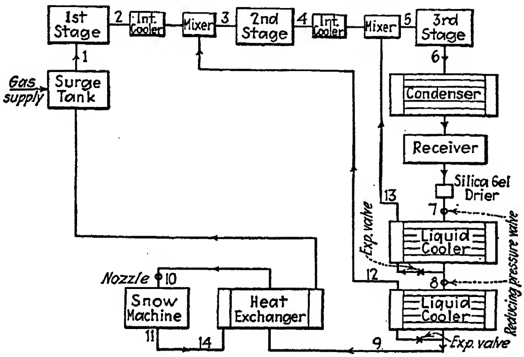


FIG. 279.—Regenerative cycle for producing carbon-dioxide snow.

A receiver stores the liquid refrigerant, which flows from it into a heat exchanger where the liquid refrigerant is cooled by the cold carbon-dioxide vapor leaving the snow machine. The carbon-dioxide gas on leaving the heat exchanger, then, enters a surge tank or mixer where it mixes with the "make-up" carbon dioxide from the gas holder. The mixed gas then enters the first compression stage completing the cycle. This is not an efficient method of producing solid carbon dioxide.

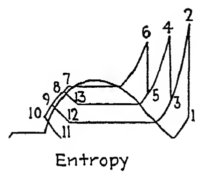


FIG. 280.—Temperature-entropy diagram for regenerative cycle shown in Fig. 279.

The *regenerative cycle* is illustrated by the arrangement shown in Fig. 279. The temperature-entropy diagram for this cycle is shown in Fig. 280. The first stage of compression is along the line 1–2; the second stage of compression is from 3–4; the third stage of compression is from 5–6. As is indicated by the constant-entropy lines, the compression is considered adiabatic. Along the paths 2–3, 4–5, the vapor is being cooled in the intercoolers. The mixers located between the first and second stages,

and the second and third stages, are to introduce the carbon-dioxide vapor that is formed by cooling the liquid refrigerant in the liquid coolers at their respective pressure levels. The carbon-dioxide vapor on being discharged into the condenser is liquefied flowing into the receiver from which it enters a silica-gel drier; it now flows to a pressure-reducing valve and into the first liquid cooler. Some of the liquid passes through an expansion valve into the cooler where it evaporates at a reduced pressure, thus cooling the liquid flowing through the first liquid cooler. In the type of cooler used here, the liquid carbon dioxide that is being cooled is separated by a metal wall from the evaporating carbon dioxide. The liquefaction is taking place along the line 6–7, and heat is removed from the liquid along 7–8 in the first liquid cooler, while more heat is removed along the line 8–9 in the second liquid cooler. The final cooling of the liquid is accomplished in the heat exchanger and occurs along the line 9–10, the cold carbon-dioxide vapor from the snow machine removing the heat. The expansion of the liquid through the nozzle in the snow chamber occurs along the line 10–11. The heating of the return or blow-back gas in the heat exchanger and mixer occurs along the line 14–1, thus completing the cycle.

The *complete reexpansion cycle* in arrangement resembles closely the regenerative cycle just described, except that the liquid coolers are replaced by flash coolers. These coolers have no metal walls, and therefore the cooling is accomplished by the pressure drop accompanied by evaporation at the pressure established in the cooler. The liquid refrigerant flowing from the liquid receiver passes through an expansion valve on entering the cooler. The gas formed in the first cooler is returned to the mixer ahead of the third compression stage. The liquid leaving the first cooler now passes through a second expansion valve before entering the second cooler. The cooling in this second cooler as in the first is accomplished by evaporation at the pressure of the cooler. The vapor thus formed is returned to the mixer ahead of the second compression stage. The liquid having been cooled in these coolers is now fed through the heat exchanger from which it enters the nozzle of the snow machine.

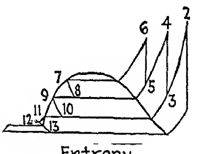


FIG. 281.—Temperature-entropy diagram for complete reexpansion cycle for making carbon-dioxide snow.

The blow-back vapor passes through the heat exchanger from which it returns to the mixer where the make-up gas is supplied.

The complete reëxpansion cycle is shown on the temperature-entropy diagram in Fig. 281, and the various lines are indicated as follows: 1–2, first-stage compression; 2–3, intercooling; 3–4, second-stage compression; 4–5, intercooling; 5–6, third-stage compression; 6–7, condensation; 7–8, liquid expansion in first cooler; 9–10, liquid expansion in second cooler; 11–12, cooling liquid in heat exchanger; 12–13, expansion through nozzle in snow chamber; 14–1, gas heating in heat exchanger and mixer.

The following data give the various pressures, temperatures, and weights for a regenerative cycle, such as shown in Fig. 279:

Pressure, pounds per square inch gage:

First stage, suction	5
First stage, discharge	75
Second stage, suction	75
Second stage, discharge	310
Third stage, suction	310
Third stage, discharge	1010

Temperature, degrees Fahrenheit:

First and second intercooler	
Water inlet	60
Water outlet	70
First stage, suction vapor	40
First stage, discharge vapor	247
Second stage, suction vapor	46
Second stage, discharge vapor	220
Third stage, suction vapor	45
Third stage, discharge vapor	198
Liquid from condenser	80
Liquid to second-stage liquid cooler	17
Liquid from second-stage liquid cooler	−40
Liquid from heat exchanger	−50
Vapor from first-stage liquid cooler	36
Vapor from second-stage liquid cooler	−62
Vapor from snow machine	−109
Vapor from heat exchanger to surge tank	−69
Vapor from make-up supply	80

Refrigerant Weight Balance, pounds:

Liquid from receiver	4.000
Vapor from first-stage liquid cooler	1.666
Liquid to second-stage liquid cooler	2.334
Vapor from second-stage liquid cooler	0.481
Liquid to snow machine	1.853

Solid carbon dioxide formed	1.000
Vapor from snow machine	0.857
Make-up carbon dioxide	1.000
Vapor to first stage	1.853
Vapor to second stage	2.334
Vapor to third stage	4.000

The following is a description of a solid-carbon-dioxide plant as arranged by the York Ice Machinery Corp. This system is somewhat different than those previously described, inasmuch as two separate refrigerating systems are used in conjunction with the compression and solidification carbon-dioxide cycle.

Carbon-dioxide Generating Side.—The common method of producing and purifying carbon dioxide is shown schematically in Fig. 282. A good grade of coke such as "seventy-two hour"

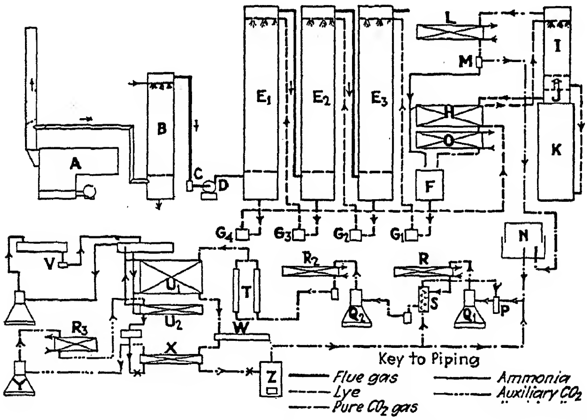


Fig. 282.—Equipment for producing and purifying carbon dioxide.

foundry coke has been found to be the best fuel for this purpose. The coke is burned in the *boiler A*, and the products of combustion are then drawn through the *scrubber B* by the induced-draft fan *D*. In large plants a forced-draft fan supplies air to the ashpit in order to maintain a balanced draft in the fire box. The *scrubber* consists of a vertical tower filled with limestone or coke which is supported by a grate, thus providing a large surface over which the water flows which is sprayed from the top. The gas entering at the bottom ascends, thus facilitating the removal of the sulphur dioxide which is absorbed by the water. It should be noted that the flue-gas constituents are carbon dioxide,

carbon monoxide, oxygen, water vapor, sulphur dioxide, and nitrogen; and the flue-gas analysis should show a high percentage of carbon dioxide in order to operate economically. The gas is cooled in the process of scrubbing, and as it leaves the scrubber it carries with it some moisture which is removed in the trap C.

The washed flue gas, which contains about 16 per cent carbon dioxide by volume, is now forced by the blower D upward through the *absorber towers* E which are in series. These absorber towers are packed with seventy-two hour coke which is supported at proper intervals by grates which prevent crushing the coke owing to its own weight. A solution of potassium carbonate is drawn from the *lye tank* F by the weak-lye pump G_1 and forced to the top of tower E_3; then through a perforated ring over the coke filling. The solution at the bottom of the tower is then pumped by means of the second weak-lye pump G_2 to tower E_2, and then by means of weak-lye pump G_3 to tower E_1. During its travel through the towers the potassium-carbonate solution has been flowing countercurrent to the flue gas, and, owing to the large surface produced by the filling, there has been brought about a very intimate contact between the two. The carbon dioxide in the flue gas reacts with potassium carbonate and produces potassium bicarbonate. The lower the temperature, the more complete the reaction, and, consequently, the larger the portion of the carbon dioxide in the flue gas absorbed by the lye solution while passing through the towers. As none of the other flue-gas constituents reacts with the lye, the inert residue of the flue gas consisting chiefly of nitrogen passes through the towers and is then discharged to the atmosphere from the top of the final tower E_3.

Thus the entire production of carbon dioxide leaves the bottom of absorber tower E_1 in the form of potassium bicarbonate. The reaction which has just taken place is reversible; that is, whereas the two combined at a comparatively low temperature to form potassium bicarbonate, if the bicarbonate solution is now sufficiently heated it will give off carbon dioxide and return to the simple carbonate form. The potassium carbonate should now be cooled thus making it available for absorbing more gas.

The bicarbonate solution is now pumped from the bottom of the absorber tower E_1 by means of the strong-lye pump G_4 and then passes through the *lye heat exchanger* H, where it is heated by the weak lye returning to the lye tank. The solution is sprayed

into the *analyzer* I, which is packed with coke. From here it trickles into the chamber J, flowing counter-current (p. 38) to the hot carbon-dioxide gas which is continually being given off in the *lye boiler* K from which the solution absorbs more heat. From the bottom of the analyzer, the solution flows by gravity into the lower end of the lye boiler K and passing upward through the tubes it is heated by steam at a pressure of 10 pounds per square inch gage. The steam condenses in the shell of the boiler. Now the potassium-bicarbonate solution changes back to potassium carbonate, liberating pure carbon-dioxide gas mixed with a considerable amount of water vapor. The carbon-dioxide gas leaves the boiler through a vertical pipe in the top head, which is protected by a hood which prevents the falling lye from entering at this outlet. Now, as the gas continues upward through the analyzer it is cooled slightly by the incoming solution. The carbon dioxide on leaving the top of the analyzer flows to the gas holder N through a water-cooled double-pipe cooler L which is equipped with a trap M to collect the condensed water vapor which drains back into the lye tank, thus maintaining the concentration of the lye solution.

From the top of the lye boiler the carbonate or weak lye solution flows by gravity into the *lye tank,* but it is first cooled by a heat exchange with the strong lye as it comes from the absorber towers in the *lye exchanger*. Its temperature is now further lowered in the water-cooled lye cooler O. The weak-lye solution is now ready to go through the cycle again.

It has been noted that a certain amount of low-pressure steam is needed in this cycle to dissociate the carbon dioxide from the bicarbonate solution in the lye boiler. This makes the operation of the compressors by steam engines very desirable, and by such an arrangement a very good heat balance is obtained between the steam and the carbon-dioxide gas generated by the coke. When using this method, the boilers are operated at a pressure of about 150 pounds per square inch gage and 100° F. superheat. The engines are designed to exhaust at a pressure of 10 pounds per square inch gage into the lye boiler. A direct connection to the steam boiler through a regulating and reducing valve is also provided to insure a full supply of steam to the lye boilers.

Compression Side, Using Separate Refrigerating Systems.—The pure carbon-dioxide gas coming from the gas holder passes through the low-pressure suction trap P before entering

stage compressor Q_1 where it is compressed to about 90 pounds per square inch gage pressure. The gas is then cooled in the *first-stage intercooler* R and then is circulated through the *dehumidifier* S which is cooled by the cold flash gas returning from the snow press (p. 448). The *second-stage compressor* Q_2 discharges into the *gas cooler* R_2 at about 500 pounds per square inch gage pressure. The gas before condensing is passed through a shell T charged with lump calcium chloride which will remove practically all of the moisture. The carbon-dioxide gas is condensed in the insulated shell and tube *condenser* U_1 at about 500 pounds per square inch pressure. This condenser is cooled by an ammonia refrigerating cycle V which operates at about 20 pounds per square inch suction-gage pressure using the shell as an evaporator.

After condensing, the liquid carbon dioxide flows into the *insulated receiver* W from which it flows to the *liquid cooler* X where it is subcooled to about $-40°$ F. The refrigeration needed for this process is furnished by an auxiliary carbon-dioxide refrigerating cycle using the cooler as an evaporator at about 90 pounds per square inch gage pressure. The *compressor* Y discharges at about 500 pounds per square inch gage pressure through a water-cooled *gas cooler* R_3 into a condenser U_2 cooled by the same ammonia cycle which is used for condensing in the primary cycle.

The subcooled liquid carbon dioxide is expanded into the *snow press* Z to a pressure slightly higher than atmospheric, and approximately 50 per cent of the liquid changes to solid carbon dioxide. The remainder of the liquid flashes into gas which returns to the low-pressure suction trap through the dehumidifier. During the process of pressing the snow into blocks and due to leakage in the press, considerable solid carbon dioxide returns to the gaseous state so that the net yield is about one-third of the liquid expanded. The remaining two-thirds, being gas, is recompressed. Thus the system actually liquefies three times as much carbon dioxide as it produces in the form of solid carbon dioxide. Since two-thirds of all the gas is always being compressed, we obtain, for every pound of gas drawn from the gas holder, a pound of solid carbon dioxide, neglecting minor losses due to evaporation.

Steam Boiler.—A boiler of the locomotive type that is well designed and suitable for burning a low-volatile fuel may be used for producing carbon dioxide. All joints, particularly those around the smoke-box doors which are asbestos packed

must be tight in order to prevent infiltration of air and a resultant dilution of the carbon-dioxide concentration of the flue gas. The fire box is water-cooled, thus making a superior type of construction to the refractory-lined furnace, where the intense heat produced by burning coke would necessitate frequent replacement of the fire brick.

Absorber Towers.—The grates which are used to support the coke filling are located about every 15 feet through the height of each tower. In order to facilitate the changing of the coke, manholes are located above and below each grate. A change in the coke filling once every 3 or 4 years is necessary. At the top of each tower a perforated ring is located so as properly to distribute the lye solution throughout the coke filling.

Lye Boiler and Analyzer.—A shell-and-tube vertical lye boiler is used for heating the lye. It is constructed like the standard vertical ammonia condenser (p. 46) with a head welded to each end, thus providing a closed circuit for the lye solution. In the upper header is welded a vertical pipe of large diameter for carrying off the pure carbon dioxide that is released in the lye boiler. This pipe has at its upper end a conical hood to prevent the lye solution from dropping directly from the analyzer into the lye boiler. In the *analyzer* a grate supports the coke filling over which perforated rings are placed similar to the arrangement used in the absorber towers.

Lye Pumps.—It is essential to design the lye pumps to possess characteristics which are favorable for operating with a throttled discharge. The double-suction volute type of pump is used and has iron fittings as lye solution attacks bronze.

Compressors.—In a plant of this type it will be apparent that it is necessary to make some departures from the standard sizes of compressors in order properly to balance the displacements.

Snow Press.—The snow press, shown in Fig. 283, in which the carbon-dioxide snow is formed and compressed into blocks is of the vertical single-acting type. The piston rod passes through the stuffing box located in the upper cylinder head and is connected to a yoke at its center. The ends of the yoke are connected to the hydraulic pistons moving in the cylinders located at each side of the snow cylinder. When the hydraulic pressure is applied, the hydraulic pistons move downward, transmitting a force to the piston in the snow cylinder and compressing the carbon-dioxide snow into blocks.

The bottom end of the snow-press cylinder is open, but when the press is to be filled with snow this opening is closed by a hydraulically operated door which is forced upward by the pressure produced in a cylinder extending down into the foundation. The door is square and of such dimensions as to cover the bottom face of the cylinder. It is made in two sections, the center one being of such size that it will fit into the press cylinder. The two sections are held together by a constant hydraulic pressure of about 500 pounds per square inch gage, and during the initial part of the pressing operation, when both the pistons and the center section of the door are subjected to a pressure of 200 pounds per square inch gage, the doors merely serve as a covering for the bottom end of the cylinder. In order to finish the operation, a hydraulic pressure of 1,200 pounds per square inch gage is applied to both the piston and the door. The result of this pressure is to force the central section of the door up into the cylinder and squeeze the lower end of the partially formed block to make it of equal density at both ends.

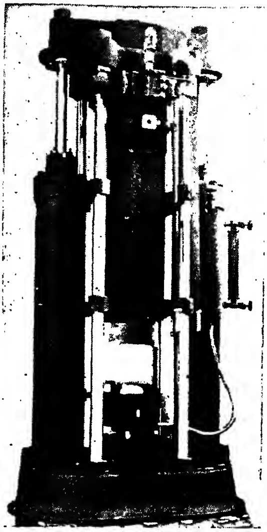


FIG. 283.—Press for making blocks of carbon-dioxide snow.

The hydraulic pressure required by the press is produced by a low-pressure pump which discharges continuously into a vertical air-cushion type of receiver or surge chamber. The low-pressure water is obtained directly from this receiver as it is needed, and this low-pressure water is used to produce the high pressure through an intensifier. A small air compressor is used to maintain the air cushion in the surge chamber.

Applications of Dry Ice.—The ice-cream industry was the first to adopt solid carbon dioxide as a refrigerant. But even in this field there is still room for much development and growth. Many manufacturers are just becoming cognizant of the superiority of this use of carbon dioxide as compared to the older methods of refrigeration. Some of its advantages are as follows: (1) cost of operation of delivery trucks is reduced tremendously; (2) truck of a given size can carry a much greater pay load than with

other methods of refrigeration; (3) truck maintenance is reduced to a minimum as solid carbon dioxide is clean and dry; (4) packing and loading are accomplished quickly; (5) shipments to longer distances are possible.

The ice-cream industry has found that fancy molds can be frozen quickly and economically by solid carbon dioxide. This refrigerant has made possible the vending of ice cream in a sanitary manner in public places and at large gatherings. At present in the Middle West large quantities of fresh meat are being transported by truck by this means of refrigeration with very good results. For the transportation of frozen foods the large distributors of frozen meat and fish have adopted it for truck deliveries. Quick-frozen fruits are also being shipped by this means.

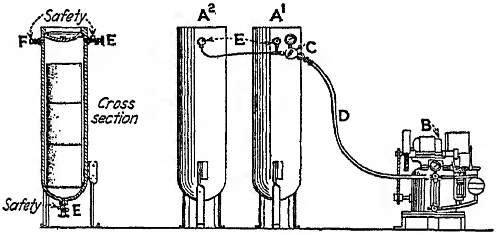


FIG. 284.—Carbon-dioxide liquefier.

As frozen foods must be kept in a hard-frozen state until used, the problem of dispensing is quite similar to that of the ice-cream industry (p. 423).

Carbon dioxide which has long been used for charging soft drinks is usually shipped in the liquid form and under high pressure. This method of shipment is quite expensive as the containers are heavy and must be shipped two ways. On the other hand, the turnover of these drums is quite small, and the fixed charges constitute a large item in the cost of producing liquid carbon dioxide. While it is recognized that the cost of solid carbon dioxide is greater than liquid carbon dioxide, the net cost to the consumer is lower. Since the two fields are generally associated, and the quality of the gas is the same, the solid carbon dioxide will serve just as well for the carbonation of beverages. In order to use solid carbon dioxide for this purpose, the bottler requires only a means for converting the solid into a liquid.

The apparatus used for coverting solid carbon dioxide into liquid carbon dioxide is known as a *liquefier* and is shown in Fig.

284. The top head has an opening which allows the solid carbon dioxide to be introduced. The liquefier should have a relief valve set at about 1,500 pounds per square inch gage pressure. The shell is equipped with a stop valve in the gas outlet and a purge valve to reduce the pressure in the liquefier when the charge is exhausted to permit the cover to be removed. Gas may be discharged from the liquefier at 100 pounds per square inch gage pressure ½ hour after it is filled. While the solid carbon dioxide is being placed in the liquefier it is, of course, giving off carbon-dioxide gas. This gas being heavier than air will fill the shell and displace any air which would interfere with the carbonating process.

CHAPTER XVI

AIR CONDITIONING

Air conditioning is the application of mechanical equipment for the removal of dust and dirt, the addition to, or the extraction of, moisture and heat from the air supplied to a building. Clean air is desirable for breathing and is sometimes essential for industrial purposes. The average person takes into his lungs more than 240 cubic feet of air in an 8-hour day. The dust in this air coats the membranes of the respiratory system with a film which seriously affects their healthy functioning, and, in addition, the particles of dust serve as carriers for germs. In the manufacture of food products, drugs, and other chemicals, such dust is extremely dangerous, and in the finishing of automobiles and furniture, and many other manufacturing processes, the elimination of dust is essential. In the air cooling of electric generators, dust must be eliminated as it clogs the narrow air passages in the generator windings and may cause serious overheating or disastrous fires.

In addition to removing the dust from the air, it is necessary to control the humidity of the air used for ventilation or manufacturing purposes. It has long been realized that humidity is an important factor from the standpoint of one's comfort, and in recent years manufacturers have learned that it also seriously affects many manufacturing processes. Theaters, restaurants, and department stores have learned that air conditioning increases their revenue during the summer months, and large offices and banks have found that it increases the efficiency of their employees during the hot weather. Manufacturers of candy, rayon, films, and many other products find that air conditioning is necessary for the manufacture of their products, and even in blast furnaces the use of properly conditioned air has been found to improve the quality and to reduce the cost of manufacture. During the cold weather, a room often feels uncomfortable even though the thermometer indicates what is normally considered a comfortable temperature. This condition is due to a lack of sufficient mois-

ture in the air, or, in scientific terms, the relative humidity is too low.

Humidity of Air.—Humidity is the moisture or water vapor mixed with the air in the atmosphere, and the weight of water vapor a given space will hold is dependent entirely on the temperature.[1] The amount of vapor in any given space is independent of the presence of the air, the only effect the air has being due to its temperature.

Absolute Humidity.—Absolute humidity is the weight of water vapor for a given volume or weight at a given temperature and percentage of saturation and is usually expressed as grains per cubic foot or grains per pound.

Relative Humidity.—Relative humidity is the ratio of the weight of water vapor in a given space to the weight which the same space is capable of containing when fully saturated at the same temperature. It is the ratio of the absolute humidity for the given condition to the absolute humidity at saturation. The quantity of moisture mixed with the air under different conditions of temperature and saturation is usually determined by means of some form of instrument in which a dry-bulb and wet-bulb thermometer are used.

Dew Point.—The dew point is the temperature at which saturation is obtained for a given weight of water vapor, or the point where any reduction in temperature would cause condensation of some of the water vapor. Any given amount of moisture must have some temperature at which saturation will occur, and any further lowering of the temperature will cause condensation. This, then, will be its dew point.

Dry-bulb and Wet-bulb Thermometer.—Usually the temperature of the air is determined by means of an ordinary or dry-bulb thermometer. The wet-bulb thermometer has the bulb covered with a piece of clean soft cloth and should be wet or dipped in water before taking a reading. Care should be always taken to keep the cloth free from dirt and to use clean pure water. This thermometer will give a depressed or lower reading than that of the dry-bulb thermometer in proportion to the evaporation from the wet surface of the cloth, and this depression is a measure of the amount of moisture in the air. This depressed reading corresponds to the temperature at which the air would

[1] Paragraphs on Humidity, Dew Point, Heat, etc., are from Section 1 of "Fan Engineering" by the Buffalo Forge Company.

normally saturate without any change in its heat contents; that is, the total heat in the air remains constant at a constant wet-bulb temperature. In order to obtain a true reading it is necessary that the thermometer be placed in a strong current of air.[1]

Sling Psychrometer.—This instrument consists of a wet- and a dry-bulb thermometer mounted on a strip of metal and provided with a handle which permits the thermometer to be rapidly whirled through the air. When being used, the instrument should be whirled continuously until no further drop in the wet-bulb reading is noted. The difference between the readings of the two thermometers is the wet-bulb depression, and, by referring to the chart on Fig. 285 of this assignment, the corresponding psychrometric conditions may be determined. There are other forms of instruments, generally of some stationary type, used for taking humidity readings, but the instrument described is reasonably accurate and is the one used by the U. S. Weather Bureau.

Relation of Dry-bulb, Wet-bulb and Dew Point Temperatures.—Dew point, as previously stated, is the temperature at which saturation is obtained for a given amount of water vapor. In other words, the air is at the dew point when it contains all the moisture it will hold at a given temperature and when it is impossible to get the air to absorb more water vapor without raising the temperature. When air has been reduced to the dew point, both wet- and dry-bulb thermometers register exactly the same; for instance, air at a temperature of 50° F. and 100 per cent saturation will contain 53.4 grains of moisture per pound, and at this condition both the dry-bulb and the wet-bulb thermometers will register 50° F. If this air is heated, both thermometers will rise, but the wet-bulb will rise more slowly than the dry-bulb temperature, and the relative humidity will be rapidly reduced. The dew point remains constant at 50° F.

[1] There is a small radiation error in the observation of the true wet-bulb reading by the wet-bulb thermometer. This is negligible for practical engineering purposes where the usual sling psychrometer is used and a vigorous whirling velocity maintained. For very low velocities, as in natural convection currents of air, the wet-bulb thermometer will not drop to the true saturation or wet-bulb temperature of the air. In order to minimize the wet-bulb error to a point where it may be entirely neglected for ordinary engineering work, some form of sling or aspiration psychrometer should be used for temperatures in still air.

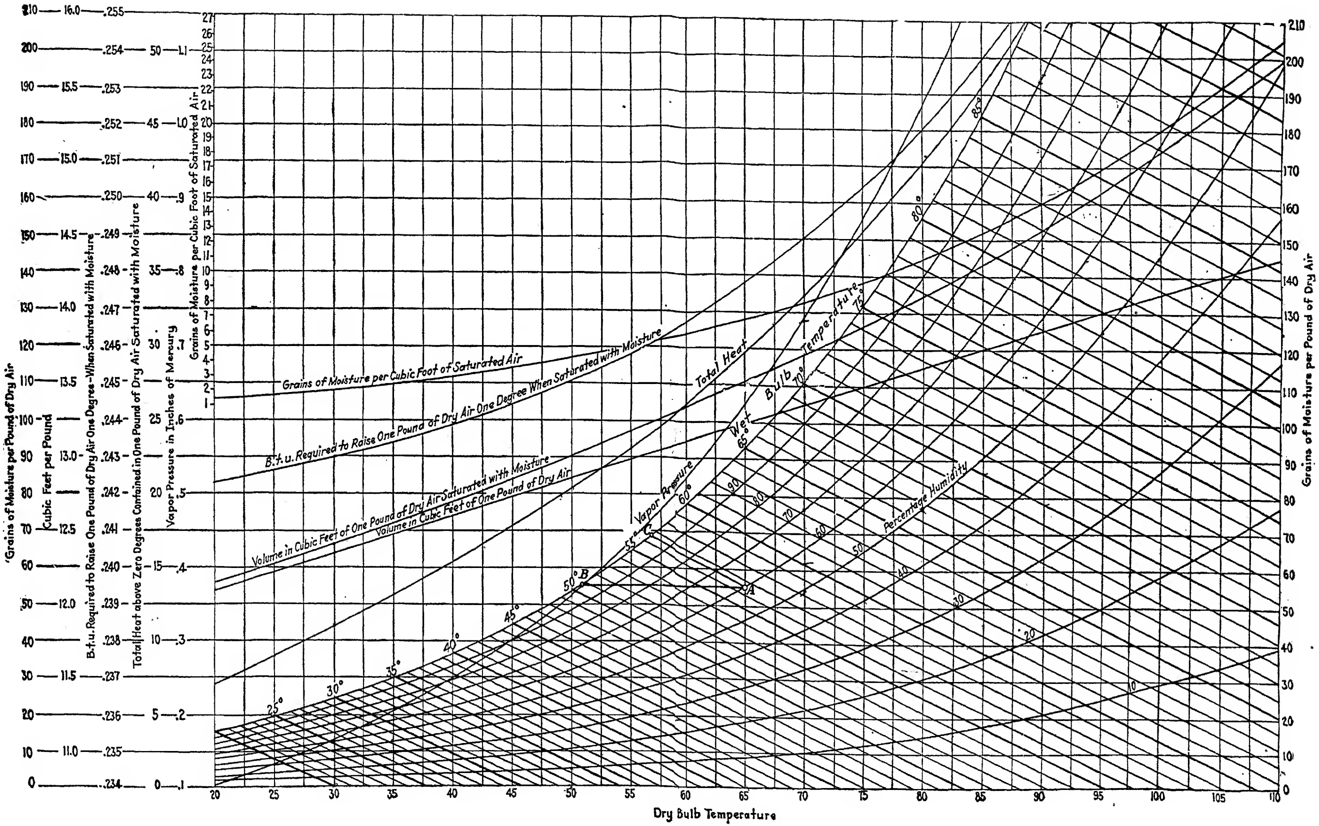


FIG. 285.—Psychrometric chart in terms of pounds of air.

(Facing page 454)

since any given number of grains of moisture per pound has a fixed and definite dew point or temperature of saturation.

If a pound of air at a temperature of 72° F. containing 53.4 grains per pound with the wet-bulb temperature at 59° F. is passed through a fine spray of recirculated water, it will absorb moisture; the dry-bulb temperature will immediately begin to fall, but the wet-bulb temperature will remain absolutely constant at 59° F., until the dry-bulb temperature has dropped to the wet-bulb temperature, namely, 59° F. As the absorption of moisture by the air takes place, the dew point will be gradually rising from 50 to 59° F. when saturation is obtained. At ordi-

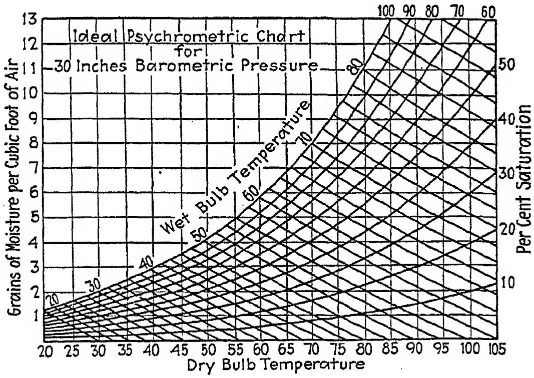


Fig. 286.—Psychrometric chart in terms of cubic feet of air.

nary temperatures, the absorption of 1 grain of moisture per cubic foot lowers the dry-bulb temperature approximately 8½° F.

Psychrometric Charts.—Charts and tables giving the important properties of air are called psychrometric.[1] The chart of Fig. 285 should be used when calculations are being made in terms of pounds of air, and the chart of Fig. 286 should be used when calculations are being made in terms of cubic feet of air.

The various curves shown on these charts will be found especially valuable in making air calculations. The grains of moisture per pound of dry air are read by following a horizontal line directly from the dew point, or intersection of the wet- and dry-bulb temperatures, to the scale on the left edge of the chart.

[1] Psychrometric charts have been prepared by W. H. Carrier and published in his paper entitled "Rational Psychrometric Formulae," which was presented before the American Society of Mechanical Engineers in 1911.

The British thermal units[1] (B.t.u.) required to raise 1 pound of dry air 1° when saturated with moisture, as also the vapor pressure, may be determined by following a vertical line from the dew point to the proper curve, and then a horizontal line to the corresponding scale on the left edge of the chart. The total heat in British thermal units (B.t.u.) above 0° F. contained in 1 pound of dry air saturated with moisture may be found by following a vertical line from the wet-bulb temperature to the total heat curve, and then a horizontal line to the left edge of the chart. The volume of air in cubic feet per pound may be found by following a vertical line from the dry-bulb temperature to either of the two volume curves and then a horizontal line to the left edge of the chart. One curve gives the volume of dry and the other of saturated air.

Example.—As an example of the use of the chart of Fig. 285 we will assume air at 75° F. dry-bulb temperature and 60 per cent relative humidity. From the chart we find that the wet-bulb temperature will be 65.25° F., the dew point is 60° F., the grains of moisture per pound of dry air are 77; the heat required to raise 1 pound of dry air saturated at 60° F. through 1° F. is 0.24664 B.t.u., and the vapor pressure of air saturated at 60° F. is 0.523 inch of mercury. By following a vertical line to the proper curve and from there a horizontal line to the scale on the left, the total heat above 0° F. in 1 pound of dry air when saturated at 65.25° F. is 29.75 B.t.u. This, then is also the measure of the heat in a pound of air at 75° F. and 60 per cent relative humidity, since the wet-bulb temperature is the same.

The cubic feet per pound of air may be found from the chart by following a vertical line from the dry-bulb temperature to either of the two volume curves of Fig. 285, depending on whether the volume of dry or of saturated air is desired. To determine the volume of one pound of partly saturated air as here assumed, we have from the chart.

Volume in cubic feet per pound at 75° F., saturated	= 13.88
Volume in cubic feet per pound at 75° F., dry	= 13.48
Volume in cubic feet of moisture per pound of saturated air at 75° F.	= .40

In this case the relative humidity is 0.60, so that the volume of moisture in cubic feet per pound at this temperature and relative humidity is 0.60 × 0.40 or 0.24 cubic foot. Volume per pound at 75° F. and 60 per cent relative humidity is 13.48 + 0.24 or 13.72 cubic feet.

Heat and Temperature.—Heat, as usually mentioned, refers to the amount of heat energy in an object, while temperature is the degree or intensity of the energy. The quantity of heat per

[1] A British thermal unit is defined on p. 95 as the quantity of heat required to raise the temperature of 1 pound of water from 62 to 63° F.

pound of the material is ordinarily expressed in terms of B.t.u. The quantity of heat within a substance depends not only on its temperature but also on its chemical composition. A pound of water at a given temperature, for example, contains more heat than a pound of air at the same temperature, due to the inherent difference in molecular structure between the two fluids. Intensity of heat is that property which causes heat to flow, or causes one body to be reduced in temperature and another one increased. The intensity of heat is ordinarily measured by the dry-bulb thermometer.

Sensible Heat of Air.—Sensible heat is the heat which may be determined by the ordinary dry-bulb thermometer and the specific heat of the material. For pure dry air

$$h = C_{pa}(t_2 - t_1)$$

where h = sensible heat in B.t.u. per pound

C_{pa} = mean specific heat = (approx.) 0.241 for air between 0 and 200° F.

$t_2 - t_1$ = increase in dry-bulb temperature of the air in degrees Fahrenheit

Sensible heat occurs in water vapor, only above saturation temperatures; that is, sensible heat of water vapor is a measure of superheat. The above equation may be used for water vapor by substituting the mean specific heat of water vapor for that of air.

Latent Heat.—In changing water into steam, heat must be added although the temperature at constant pressure is not changed. Thus, at atmospheric pressure, water at 212° F. is vaporized at the same temperature by the addition of heat at the rate of 970.4 B.t.u. per pound. The heat required to perform this change of state is termed *latent heat,* and its amount varies according to the temperature. Likewise it requires heat to evaporate water at temperatures below the boiling point; and as this heat must be obtained from some source, the air which is in direct contact with the vapor will exhibit the well-known reduction in temperature, wet-bulb depression, to furnish this latent heat of vaporization. The latent heat of water vapor may be calculated for temperatures between 40 and 150° F. from the approximate formula $L = 1091.6 - 0.56t$ (see p. 458).

Total Heat of Air.—As ordinary air contains water vapor in varying amounts, the *latent* heat of this vapor must be added to the *sensible* heat of the air mixture (air and water vapor) in order to obtain the *total heat* of the mixture. The total heat is a con-

stant quantity for any given wet-bulb temperature, irrespective of any change in the dry-bulb temperature. This fact has been termed by W. H. Carrier "one of the four fundamental psychrometric principles," meaning that the true wet-bulb temperature of the air depends entirely on the total amount of the sensible and the latent heats in the air and is independent of their relative proportions. In other words, the wet-bulb temperature of the air is constant, providing the total heat of the air is constant. Thus, if sufficient moisture is introduced into a certain quantity of air, the dry-bulb temperature of the air will be lowered until it is the same as the wet-bulb temperature. This is simply an exchange from sensible heat into latent heat required to vaporize the moisture, if the total heat is kept the same. Since no heat passes either into or out of the system, the process has been referred to as *adiabatic saturation* of the air. If a further lowering of the temperature takes place, the wet-bulb temperature will be lower, and the corresponding total heat will be less. If the air should be heated without the addition of more moisture, the dew-point temperature of the air would remain constant, but the wet-bulb, as well as the dry-bulb, temperature would increase, and the total heat of the air would increase a corresponding amount. The psychrometric chart of Fig. 285 will be found especially convenient in following these changes in heat content. The total heat may be obtained from the curve of total heat, as already explained.

It will be observed that the values of total heat above 0° F. may be most readily calculated for saturated air by the formula:

$$H = tC_{pa} + Lw$$

where H = total heat, above 0 degrees Fahrenheit in B.t.u. per pound

C_{pa} = specific heat of air at temperature t

t = wet-bulb temperature of the air °F.

L = latent heat of vapor at wet-bulb temperature

w = weight, pounds of vapor per pound of dry air in the mixture

The total heat above 0° F. of 1 pound of saturated air at 60° F. may be calculated thus,

$t = 60$ deg. Fahr.

C_{pa} at 60 deg. Fahr. $= 0.241$

$L = 1091.6 - 0.56 \times 60$ (from formula on p. 457) $= 1{,}058$

W from chart in Fig. 286 = 77 grains

1 pound = 7,000 grains

From the formula the total heat of saturated air at 60° F. is

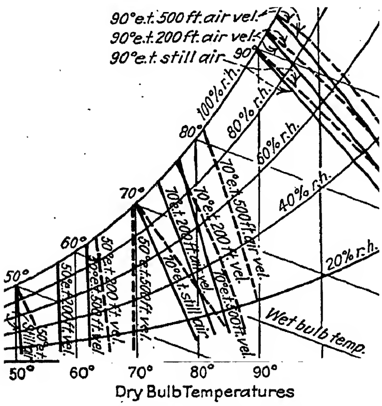
$$H = 60 \times 0.241 + \frac{1{,}058 \times}{7{,}000} = 14.46 + 11.62$$
$$= 26.08 \text{ B.t.u. per pound}$$

From the chart of Fig. 285, reading vertically upward from 60° F. on the saturation (100 per cent humidity) line to the *total heat* curve and then horizontally from the intersection with this curve to the scale of total heat, the dry air saturated with moisture at 60° F. has a total heat of 26 B.t.u. per pound.

Effective Temperature.—It is a familiar fact that a room may be comfortable at one temperature and yet decidedly uncomfortable at another time, even though the same dry-bulb temperature prevails. This may be due to a change of either relative humidity or air motion. On the other hand, the room may feel equally comfortable under different conditions of dry-bulb temperature provided either relative humidity or air motion is changed to produce this effect. The name "effective temperature" has been used to designate any series of conditions producing the same degree of comfort upon the human body. The American Society of Heating and Ventilating Engineers has, at its research laboratory, investigated and charted these conditions over a wide range of temperatures, humidity, and air motion. Original tests were made on persons stripped to the waist and in still air, while later the effect of air motion was studied. These studies have revolutionized the whole science of heating and ventilating with regard to comfort. It is practical now to obtain a degree of comfort in theaters and auditoriums in hot weather, or in industrial plants, such as glass factories and steel mills, which heretofore was thought impossible. The newer science is a simple application of fans and air washers with a properly designed distribution system.

In order to understand more clearly the value of effective temperature, the chart of Fig. 287 is given, showing three effective temperature lines for 50, 70, and 90° F. and three velocities of 0, 200, and 500 feet per minute. It will be noted that the method of naming the effective temperatures has been purely arbitrary. At saturation the dry-bulb, wet-bulb, and dew-point temperatures coincide, and this has been chosen to desig-

nate the degrees of effective temperature in still air. The solid lines are persons normally clothed while the dotted lines are for persons stripped to the waist. Thus, for a person normally clothed all conditions represented, say, by the solid 70° F. effective-temperature lines, will be equally comfortable. Likewise for a person stripped to the waist all conditions represented by the dotted 70° F. effective-temperature lines will be equally comfortable. Thus for a given air condition (dry-bulb, wet-bulb, and velocity) the difference in effective temperatures with and without clothing is a measure of heating (or cooling) effect produced by the clothes.



Note: Solid lines are effective temperatures for persons normally clothed and slightly active Dotted lines are effective temperatures for persons stripped to the waist and at rest

FIG. 287.—Effect of clothing on effective room temperatures.

Comfort Zone.—Elaborate tests have been made to determine the effective-temperature range most comfortable to the majority of people. This has been termed the comfort zone, and for persons normally clothed and slightly active may be taken as between 63 and 71° F. effective temperature. The comfort line is that effective temperature reported comfortable by the largest percentage of people tested and is found to be 66° F. For persons stripped to the waist the comfort zone lies between 61 and 69° F. effective temperature, and 64° F. may be taken as the comfort line.

Humidity in Heated Buildings.—The question of humidity in heated buildings is of more consequence than is usually supposed. When air is heated in the absence of free water it becomes "drier," although the actual amount of water vapor present remains the same. In other words, the absolute humidity is the same, but the relative humidity has been lowered. Thus, suppose that air enters the ventilating system at 20° F. and contains 10 grains of moisture per pound of air. Saturated air at this temperature contains about 15 grains per pound of air, so that the entering air is two-thirds saturated or has a relative humidity of 66 per

cent. If this same air is now heated to 70° F. without adding any moisture to it, it will still contain 10 grains of moisture per pound of air. However, saturated air at 70° F. contains about 110 grains per pound of air, so that the heated air will now be one-eleventh saturated, or its relative humidity is 9.1 per cent. It will be readily seen that the term "humidity" as used in the accustomed sense really means *relative humidity* and is the ratio of the actual amount of water vapor present to what could be present were the air saturated. Moreover, this heated air is very dry and injurious to the nose, throat, and lungs, when constantly breathed. To overcome this difficulty, moisture should be added to the air in addition to heat, so that when introduced into the building its humidity will be more nearly that suited to the needs of the human body.

A proper relative humidity to be maintained in public buildings is from 35 to 50 per cent. The relative humidity to be recommended in good practice is 40 per cent, with a room temperature of 68° F. This corresponds to about 3 grains of moisture per cubic foot of air and a dew point of 42° F. Even this will cause slight condensation on the windows in extremely cold weather, and a lower humidity should be maintained in very cold weather if condensation on the windows is objectionable.

The most practical method of adding moisture to the air and securing the desired humidity is by means of the air washer or the humidifier. The air can thus be cleaned as well as humidified. The method of controlling the temperature and humidity of the air leaving the washer or humidifier can be very closely governed by automatic regulation, as will be explained later. The average home heated with a hot-air furnace is one of the worst offenders from the viewpoint of containing too dry air. The usual water pan does not evaporate enough water for normal requirements. Recirculation of air will help much toward increasing humidity, and when a small recirculating fan is provided there will be a more rapid movement of air through the piping, and more comfortable conditions with lower temperatures at the air inlets in the rooms will be obtained. This is not only a practical but an efficient system, especially for the coldest weather conditions. In the larger residences complete installations of fans and air washers may be used.

Method of Using the Charts.—In order to follow the solution of the various examples and to solve similar problems, it is first

necessary to study and understand Fig. 286. If the readings on a sling psychrometer (p. 454) show a dry-bulb temperature of 70° F. and a wet-bulb temperature of 60° F., the relative humidity, the number of grains of moisture per pound of dry air, the total heat in B.t.u. in 1 pound of air, the dew point in degrees Fahrenheit, the number of grains of moisture per cubic foot of saturated air, and the volume in cubic feet of 1 pound of air can be calculated by the use of the chart in the figure. The point of intersection of the vertical line from the dry-bulb temperature of 70° F. and the diagonal line for the wet-bulb temperature of 60° F. is about midway between the curves for 50 and 60 per cent relative humidity. The relative humidity is, therefore, 55 per cent. From the same point a horizontal line to the scale at the extreme left gives a reading of 61 grains of moisture per pound of dry air.

A constant wet-bulb temperature represents constant total heat. If, therefore, the line of wet-bulb temperature is followed to its intersection with the curve for 100 per cent relative humidity, and from this point of intersection vertically to the point of intersection with the curve for total heat, and from there horizontally to the scale for total heat above 0°, it is shown that the air has a total heat above zero of 26 B.t.u. per pound. Following, similarly, a horizontal line from the point of intersection of the lines for wet- and dry-bulb temperatures, this line intersects the curve for 100 per cent relative humidity at a temperature of 53.5° F. This is the *dew point*, or the temperature below which the moisture in the air will condense. If from the dew point a vertical line is followed to the point of its intersection with the curve for grains of moisture per cubic foot of saturated air, and from this point of intersection horizontally to the corresponding scale, we find that the air contains 4.7 grains of moisture per cubic foot of saturated air.

To find the volume in cubic feet of 1 pound of air, first follow a vertical line from the 70° F. dry-bulb temperature to the point of intersection with the curve for volume in cubic feet of 1 pound of dry air saturated with moisture, and from this point horizontally to the corresponding scale. The volume is found to be 13.7 cubic feet, and, similarly, the volume of 1 pound of dry air is found to be 13.33 cubic feet. The volume of vapor for 100 per cent humidity must, therefore, be 13.72 − 13.33 cubic feet, or 0.39 cubic foot, and for 55 per cent humidity the volume will be

0.39×0.55, or 0.21 cubic foot. The volume of 1 cubic foot of air at 70° F. dry-bulb and 60° F. wet-bulb temperature will, therefore, be $13.33 + 0.21$ or 13.54 cubic feet.

When considering the effects of various changes in the conditions of the air, the two rules following should be remembered:

1. If no moisture is added or taken away, any change in wet- or dry-bulb temperatures, relative humidity, and total heat must be so related that *the dew point and absolute humidity remain constant.*

2. If *no heat is added or taken away,* any change in relative humidity, dew point, total humidity, or dry-bulb temperature must be so related that *the wet-bulb temperature will remain constant.*

Example 1: The outside air is at 30° F. dry-bulb temperature and 40 per cent relative humidity as supplied to a ventilation system. The air first passes through a tempering coil where it is heated to 55° F. It then passes through an air washer of such design that the air when leaving is 100 per cent saturated. The air next passes through a reheating coil where the temperature is raised to 80° F. What will be the relative humidity of the air leaving the reheating coil?

Referring to Fig. 285 and following a vertical line from 30° F. dry-bulb temperature to the intersection of this line with the curve for 40 per cent relative humidity and then horizontally to the scale of grains of moisture per pound of dry air, it is shown that 1 pound of air contains 10 grains of moisture. After leaving the tempering coil the air will be at a temperature of 55° F., and as it still contains the same total amount of moisture, the same horizontal line is followed to the point of intersection with the vertical line from 55° F. dry-bulb temperature, showing that the relative humidity is now about 15.5 per cent and the wet-bulb temperature 39° F. Passing through the air washer the air will be cooled to the wet-bulb temperature (39° F.) and will be 100 per cent saturated when leaving the washer. By following a horizontal line from the point of intersection of the 39° F. line and the line for 100 per cent relative humidity, the scale of moisture indicates that the air now contains 34 grains of moisture per pound. Following this same line to the point of intersection with the line for 80° F. dry-bulb temperature, the relative humidity of the air after leaving the reheater is 22 per cent.

Example 2: If it is assumed that the same outside air conditions prevail as in Example 1, it is desired to have the air leave the reheater at 80° F. and 35 per cent relative humidity; what should be the temperatures of the air leaving the tempering coils and the reheating coils?

Referring again to Fig. 285, we find that air at 80° F. and 35 per cent relative humidity contains 53 grains of moisture per pound of air and that the dew point (point of 100 per cent saturation) is 50° F. This must be the temperature at which the saturated air will leave the air washer and also the wet-bulb temperature of the air entering the air washer. In Example 1 the calculations showed that the outside air at 30° F. dry-bulb temperature

and 40 per cent relative humidity contains 10 grains of moisture per pound of dry air.

On the chart (Fig. 285) at the point of intersection of the horizontal line indicating 10 grains per pound and the line for 50° F. wet-bulb temperature is the dry-bulb temperature of the air entering the air washer. This temperature is 77.5° F. It is, therefore, necessary to provide tempering coils to heat the air from 30 to 78° F., and reheating coils to heat the air from 50 to 80° F.

In order to reduce the lowering in temperature of the air passing through the air washer, provision is sometimes made for heating the water supplied to the spray nozzles of the washer.

Example 3: One thousand pounds of air per minute are supplied to a building from outdoors at a temperature of 30° F. and 20 per cent relative humidity. After passing through tempering coils where it is heated to 50° F., the air passes through an air washer and then through reheating coils where it is heated to 75° F. If a final relative humidity of 40 per cent is desired, to what temperatures must the air-washer water be heated, and how much steam per hour at 5 pounds per square inch gage pressure will be required to heat the water?

Reference to the chart in Fig. 285 shows that air at 75° F. and 40 per cent relative humidity contains 52 grains of moisture per pound and that the temperature of the air leaving the washer, if 100 per cent saturated, must be 49° F. The total heat of the saturated air at 49° F. is 19.75 B.t.u. per pound. Outside air at 30° F. and 20 per cent relative humidity contains 5 grains of moisture, and when heated to 50° F. will have a relative humidity of 10 per cent and a total heat of 13 B.t.u. per pound. It is, therefore, necessary to supply 19.75 − 13 or 6.75 B.t.u. per pound of air. The total B.t.u. per hour will, therefore, be 6.75 × 1,000 × 60 or 405,000 B.t.u. per hour.

Assuming that steam at 5 pounds per square inch gage pressure will supply 961 B.t.u. then $\frac{6.75 \times 1{,}000 \times 60}{961}$ will equal 421 pounds of steam per hour, and the water must be heated to a temperature of 49° F.

Example 4: It is desired to supply 50,000 cubic feet of air per minute to a factory at a temperature of 80° F. and 50 per cent relative humidity. Outside air is at 30° F. and 40 per cent relative humidity. How much steam per hour at 5 pounds pressure and how many gallons of water per minute will be required?

From the chart in Fig. 285 the following data are obtained:

Volume of 1 pound of saturated air at 30° F. is 12.4 cubic feet.

Volume of 1 pound of dry air at 30° F. is 12.32 cubic feet.

Volume of vapor per pound of saturated air is 0.08 cubic foot.

Volume of vapor per pound of 40 per cent saturated air is 0.032 cubic foot.

Volume of 1 pound of air at 30° F. and 40 per cent relative humidity is therefore 12.32 + 0.032 or 12.35 cubic feet.

Weight of 50,000 cubic feet of air will be or 4,050 pounds.

Total heat per pound of air at 80° F. and 50 per cent relative humidity is 31 B.t.u.

Total heat per pound of air at 30° F. and 40 per cent relative humidity is 8.8 B.t.u.

Heat to be supplied per pound of air is 22.25 B.t.u.
Total B.t.u. required per hour will be $22.25 \times 4{,}050 \times 60$ or 5,406,750.
Heat available per pound of steam at 5 pounds pressure is 961 B.t.u.

Pounds of steam required $= \dfrac{22.25 \times 4{,}050 \times 60}{961}$ or 5,620 pounds per hour.

Moisture per pound of air at 80° F. and 50 per cent relative humidity is 76 grains.
Moisture per pound of air at 30° F. and 40 per cent relative humidity is 10 grains.
Moisture to be added per pound of air is 66 grains.

Gallons of water per minute will, therefore, be $\dfrac{4{,}050 \times 66}{7{,}000^{*} \times 8.3^{\dagger}}$ or 4.60 gallons.

Example 5: During the summer months the outside air at a factory location has a temperature of 100° F. and 50 per cent relative humidity. Air for ventilation of the factory is so divided that two-thirds of the air from the outside is passed through an air washer where it is saturated 100 per cent and one-third is bypassed, so that it does not go through the air washer. The air is then mixed before entering the fan supplying air for the building. What will be the resultant temperature and relative humidity of the air entering the fan?

Air at 100° F. and 50 per cent relative humidity contains 145 grains of moisture per pound, and the wet-bulb temperature is 83.2° F.

Passing through the air washer the air will become saturated and will be cooled to the wet-bulb temperature, or 83.5° F. and will contain 175 grains of moisture per pound. Mixing the air so that there are 2 pounds of air at 173 grains per pound, or 346 grains, and 1 pound of air at 145 grains per pound, or 145 grains, this makes a total of 3 pounds of air containing 491 grains, or 164 grains *per pound* of air.

The temperature of the mixed air is found as follows:

Two pounds of air at 83.5° F. are equivalent to 2×83.5 or 167 temperature units

One pound of air at 100° F. is equivalent to 100 temperature units

There is, then, a total of 3 pounds of air with the equivalent of $167 + 100$ or 267 temperature units, or 1 pound with one-third of 267 or 89 temperature Fahrenheit units. The resultant temperature of the air is, therefore, 89° F. and the relative humidity 77 per cent.

Example 6: If it is assumed in the previous example that ample cooling water is available at 60° F., what will be the resultant temperature and relative humidity? When passing through the air washer, the air will be cooled to the cold-water temperature, or 60° F. and will be saturated at that temperature. The air will, therefore, contain 78 grains of moisture per cubic foot.

Mixing the air as before there will be:

Two pounds of air at 78 grains per pound, or 156 grains
One pound of air at 145 grains per pound, or 145 grains

and a total of 3 pounds of air with 301 grains, or 100 grains per pound of air.

The temperature of the air may be determined as follows:

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* 7,000 grains per pound.

† 8.3 pounds per gallon.

Two pounds of air at 60° F. are the equivalent of 120 temperature units
One pound of air at 100° F. is the equivalent of 100 temperature units
There is a total of 3 pounds of air with 220 temperature units, or 1 pound at 73° F. The temperature of the air will, therefore, be 73° F. and the relative humidity will be 82 per cent.

By reducing the proportion of air passing through the air washer and increasing the proportion bypassed, the relative humidity may be decreased, but the temperature will be increased.

Example 7: It is desired to supply 10,000 cubic feet of air per minute to a theater at 85° F. when the outside air has a temperature of 100° F. and 60 per cent relative humidity. Part of the air will be bypassed, and the balance will pass through an air washer supplied with ample cold water at 50° F. and of suitable design so that the air passing through it will be 100 per cent saturated. What proportion of the air must pass through the air washer, and what will be the relative humidity of the air in the theater?

Assume that X cubic feet of air per minute pass through the air washer. Then $10{,}000 - X$ cubic feet per minute will be bypassed. Mixing X cubic feet per minute at 50° F. and $(10{,}000 - X)$ cubic feet per minute at 100° F., it is required to obtain 10,000 cubic feet per minute at 85° F.; and this can be expressed by the equations:

$$50X + (10{,}000 - X)100 = 10{,}000 \times 85$$
$$50X + 1{,}000{,}000 - 100X = 150{,}000$$
$$150{,}000 = 50X$$

Then, $X = 3{,}000$ which is the number of cubic feet per minute passing through the air washer, and consequently 7,000 cubic feet per minute will be bypassed.

Three pounds of air at 50° F. and 100 per cent saturated contains 159 grains, and 7 pounds of air at 100° F. and 60 per cent relative humidity contains 1,225 grains. Then, 10 pounds of air at 85° F. will contain 159 + 1,225 or 1,384 grains, and its relative humidity will be 76 per cent.

Example 8: Fresh air for a theater is at 40° F. dry-bulb temperature and 20 per cent relative humidity. After passing through an Aerofin heater or similar radiation heater it enters the air washer at 60° F. In its passage through the air washer it becomes 85 per cent saturated. The air is then reheated to 75° F. What will be the relative humidity of the air leaving the reheating coil?

Referring to Fig. 285, follow a vertical line from the 40° F. dry-bulb temperature to the point of intersection with the curve for 20 per cent relative humidity. From this point follow a horizontal line to the scale on the left, and it will be found that the air contains 7 grains of moisture per pound of dry air. After leaving the tempering coil the air will be at a temperature of 60° F. and, as it contains the same amount of moisture, the relative humidity will now be 10 per cent and the wet-bulb temperature 41° F. When passing through the air washer the air becomes 85 per cent saturated. By following along the line for 41° F. wet-bulb temperature to its intersection with the curve for 85 per cent relative humidity, the dry-bulb temperature is 43° F., and the air contains 34 grains of moisture per pound.

The air then passing through the reheater coils is heated to 75° F. As the moisture content remains the same, the relative humidity is 27 per cent.

Example 9: Outside air at 30° F. and 50 per cent humidity is to be supplied to a room at 70° F. and 40 per cent relative humidity. If the air washer will saturate the air to 85 per cent humidity, what temperature should the air have when entering and leaving the air washer?

Air at 70° F. and 40 per cent relative humidity contains 43 grains of moisture per pound. The same amount of moisture will be contained in the air when leaving the washer and, as the relative humidity is 85 per cent, the dry-bulb temperature must be 49° F. and the wet-bulb temperature 47° F. Air at 30° F. and 50 per cent relative humidity contains 12 grains of moisture per pound and, as it will contain the same amount when entering the air washer, the dry-bulb temperature of the air entering the washer must be 69° F. Therefore, the tempering coils should be selected to heat the air from 30° F. to 69° F., and the reheating coils should be designed to heat the air from 49 to 70° F.

Example 10: It is desired to maintain a temperature of 80° F. and 70 per cent relative humidity in a factory in which 12,000 B.t.u. per hour are given off by equipment and the piping in the building. The outside air is assumed to have a temperature of 80° F. and a relative humidity of 40 per cent. Assume that the air passes through an air washer of suitable design to obtain 100 per cent saturation of the air and that ample cold water is available at 50° F., how much air must be supplied to the factory per minute and to what temperature should the water be heated?

The dew point of air at 80° F. and 70 per cent relative humidity is 69° F., which should be the temperature of the water to obtain the maximum cooling effect with the desired relative humidity.

Heat content per pound of air at 80° F. and 70 per cent relative humidity is 35.5 B.t.u.

Heat content per pound of air at 69° F. and 100 per cent relative humidity is 32.5 B.t.u.

Heat absorbed per pound of air is 3 B.t.u.

Heat to be absorbed is 12,000 B.t.u. per hour or 200 B.t.u. per minute, so that it will, therefore, be necessary to supply 200 ÷ 3 or 66.6 pounds of air per minute.

Volume per cubic foot of saturated air at 69° F. is 13.65 cubic feet. The volume of the air to be supplied is 66.6 × 13.65 or 910 cubic feet per minute.

For ventilation computations it is usual to calculate in terms of cubic feet of air per minute rather than pounds of air per minute. In Fig. 286 the moisture content is, therefore, given in grains per cubic feet of air, and, as most air-conditioning computations are limited to the determination of wet- and dry-bulb temperatures, relative and absolute humidity and dew point, the curves and scales for total heat, volume, etc., are omitted from this chart.

Example 11: Ten thousand cubic feet of air per minute is to be supplied to a theater at 70° F. and 35 per cent relative humidity, with an outside temperature of 20° F. and a relative humidity of 50 per cent. How many gallons of water per hour must be evaporated?

From Fig. 286 we find that 1 cubic foot of air at 70° F. and 35 per cent relative humidity contains 2.75 grains, and that 1 cubic foot of air at 20° F. and 50 per cent relative humidity contains 0.6 grain, so that the moisture

to be added per cubic foot of air is 2.15 grains. Then, the gallons of water required per hour will be $\frac{10{,}000 \times 60 \times 2.15}{7{,}000 \times 8.3}$ or 22.2 gallons per hour.

Example 12: If 40,000 cubic feet of air per minute are supplied to a generator room with the outside air at a temperature of 100° F. and 40 per cent relative humidity, how many cubic feet per minute should pass through an air washer designed to saturate the air 100 per cent if the final temperature is to be 90° F., and what will be the relative humidity?

Assuming that X cubic feet of air per minute pass through the air washer, Fig. 286 can be used to find the wet-bulb temperature which is 79.2° F. Then

$$79.2X + (40{,}000 - X)100 = 40{,}000 \times 90$$
$$79.2X + 4{,}000{,}000 - 100X = 3{,}600{,}000$$
$$20.8X = 400{,}000$$
$$X = 19{,}200 \text{ cubic feet per minute through the air washer.}$$

The relative humidity is found as follows:

19,200 cubic feet of air at 10.8 grains per cubic foot = 207,500 grains
20,800 cubic feet of air at 8 grains per cubic foot = 166,400 grains
40,000 cubic feet of air contains 373,900 grains, or 9.34 grains per cubic foot.

This corresponds to a relative humidity of 63 per cent.

Dew-point Method of Humidity Control.—In cotton mills, tobacco factories, and certain other industries it is necessary during the warm weather, and sometimes throughout the year, to cool as well as humidify the air. It is essential in such cases that the difference between the dew point of the incoming air and the room temperature shall not exceed a definite value dependent upon the humidity desired. A careful examination of the chart on Figure 285 will show that for any one percentage of humidity there is a nearly constant difference between the dry-bulb temperature and the corresponding dew point. If, for example, we consider a relative humidity of 50 per cent we find that the dry-bulb temperatures and the corresponding dew points are as given in Table XVI.

TABLE XVI.—DRY-BULB AND DEW-POINT TEMPERATURE DIFFERENCES

Dry bulb, degrees Fahrenheit	Dew point, degrees Fahrenheit	Temperature difference, degrees Fahrenheit
100	78	22
90	69	21
80	59.5	20.5
70	50	20
60	41	19
50	32	18

The minimum temperature at which air can be introduced is evidently the dew point or saturation temperature at the air washer or humidifier, for if it were introduced at a lower temperature there would be insufficient moisture to give the desired humidity at the room temperature. This permissible temperature rise limits the possible cooling effect to be obtained from each cubic foot of air as shown in the accompanying table. This relationship is of primary importance in the design of the humidifying system, and the disregard of it or failure to understand it has in many cases been the cause of failure or unsatisfactory operation.

TABLE XVII.—COOLING CAPACITY OF DEHUMIDIFICATION SYSTEM

Humidity desired in room, per cent	Difference between dew point and room temperature, at 80° F., degrees Fahrenheit	Air at 70° F. required per B.t.u. cooling effect, cubic feet
50	20.3	2.71
55	17.7	3.11
60	15 2	3.63
65	12.8	4.31
70	10.8	5.10
75	8.8	6.27
80	6.8	8.11

If a temperature of 80° F. and a relative humidity of 60 per cent are to be maintained in a factory where manufacturing equipment gives off 12,000 B.t.u. per hour and the temperature of the incoming air must not be below 70° F., the cubic feet of air required per minute and its relative humidity can be calculated.

From Table XVII the dew point of the entering air should be 80 − 15.2 or 64.8° F. At 70° F., the air must have a relative humidity of 84 per cent, and the cubic feet of air required per B.t.u. cooling effect will be 3.63. To obtain a cooling effect of 12,000 B.t.u. per hour will require $\frac{12{,}000 \times 3.63}{60}$ or 726 cubic feet of air per minute.

When the spray water is recirculated without heating, as in warm weather, it remains at all times substantially at the wet-bulb temperature of the entering air, while the wet-bulb temperature of the air leaving the washer or dehumidifier is unchanged; it follows, then, in conformance with the theory, that when the

air is completely saturated as in the humidifier, the air is cooled to the wet-bulb temperature of the incoming air. This cooling effect is due to evaporation and is, therefore, in direct proportion to the moisture added to the air. The wet-bulb depression in atmospheric air averages from 12 to 15° F. in summer, while occasionally a depression of 20 to 30° F. is found in extremely hot and dry weather. In every case a properly designed humidifier will cool the incoming air a corresponding number of degrees.

When saturation is incomplete, as in the ordinary air washer, the wet-bulb depression of the air leaving the washer is found to be a constant percentage of the initial wet-bulb depression, when the air velocity remains constant.

It follows that the cooling effect is a constant percentage of the initial wet-bulb depression. This may be expressed by the formula

where t' = constant wet-bulb temperature
t_1 = temperature of air entering washer
t_2 = temperature of air leaving washer
R = constant ratio depending upon intimacy of contact, air velocity, etc.
E = efficiency of saturation,
which is $1 - R$, so that

If the dry-bulb temperature is 80° F. and the wet-bulb temperature is 60° F., what is the efficiency of an air washer if the temperature of the air leaving the washer is 65° F.

$$\frac{80 - 65}{80 - 60} \quad \frac{15}{20} \text{ or 75 per cent}$$

In every industrial air-conditioning plant there are *four* sources of heat which must be taken into account in the design of the system:

a. Radiation from the outside air owing to the maintenance of a lower temperature inside. At ordinary humidities this is negligible, but at high humidities and in dehumidifying plants it is an important factor, due to the increased temperature difference. This may be calculated from the usual constants of radiation.

b. The heating effect of direct sunlight is especially noticeable from window shades and exposed windows and sky-lights where the entire heat energy of the sunlight is admitted to the room, and from the roof which constitutes the greater amount of sunlight exposure and which in ordinary construction transmits heat much more readily than the walls. Precautions should be taken where high humidities are desired to shade exposed windows and to insulate the roof thoroughly. Ventilators in the roof are of great advantage in removing the hot layer of air next to it, and ample capacity should always be provided in such units.

c. Radiation of heat from the bodies of the operatives in a building amounts of about 400 to 500 B.t.u. per hour per operative, a portion of which is sensible heat, the remainder being transformed into latent heat through evaporation.

d. The heat developed by the power consumed in driving machinery and in manufacturing processes in general is another item of heat supply. According to the laws of the conservation of energy, all power used in manufacturing is ultimately of energy, therefore, creates 42.42 B.t.u. of heat per minute, for which ventilation must be provided. In high-powered mills this is the chief source of heating, and this heat is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

It must be remembered that in cooling moist air the latent heat removed in condensing the moisture is usually of more importance than the reduction in the sensible heat of the air itself. The total heat removed may be determined from the total-heat curve on Fig. 285, as shown in previous examples. It should also be noted that the total heat of the air depends on the *wet-bulb temperature* only, and the *wet-bulb temperature should always be used in such calculations.*

Calculation of Air Conditioning.—In the examples which have been previously given, we have found that unless sufficient cold water is available to make it unnecessary to recirculate the water in the air washer, the degree of cooling that can be accomplished depends on the relative humidity and cannot be any greater than the wet-bulb depression. Furthermore, the benefits of lower temperature may be largely offset by the increase in humidity. In some industries an increase in humidity may be desirable or even necessary, but where comfort is essential, as in moving-picture theaters, restaurants, etc., it is desirable to provide some means to reduce both temperature and humidity, and, in addition, to absorb the heat given off by a large number of people and lights. Unless a natural source of cold water is available, artificial refrigeration must be used to provide the necessary cooling. As the cooling plants for this class of work

run into large capacities, the question of low operating expense, as well as low first cost, is of vital importance. A difference of 12 to 15° F. between the inside and outside temperatures with a wet-bulb temperature inside the building of about 70° F. should generally prove satisfactory, when the outside temperature is 90° F. or more.

Example 13: In a theater having a seating capacity of 2,000 people it is desired to maintain a temperature of 80° F. and 70 per cent relative humidity, when the outside temperature is 95° F. and the relative humidity is 50 per cent. The heat given off per person equals 300 B.t.u. per hour; the heat given off by the lights is equal to 20,000 B.t.u. per hour; the heat transfer through the walls is equal to 100,000 B.t.u. per hour, 50 per cent of the air being recirculated. What must be the capacity of the fan in cubic feet per minute, the temperature to which the water in the air washer must be cooled, and the tons of refrigeration required at the air washer to accomplish this?

Since the air leaves the theater at 80° F. and 70 per cent relative humidity, the heat absorbed by the air in its passage through the theater is $2{,}000 \times 300 + 20{,}000 + 100{,}000 = 720{,}000$ B.t.u. per hour.

As only 50 per cent of the air is recirculated, this quantity must be doubled to provide for the cooling of an equal amount of air from 95° F. and 50 per cent relative humidity to 80° F. and 70 per cent relative humidity.

Air at 95° F. and 50 per cent relative humidity contains 41.75 B.t.u. per pound, and air at 80° F. and 70 per cent relative humidity contains 35.25 B.t.u. per pound, so that the heat absorbed per pound of air equals 6.50 B.t.u. To absorb 720,000 B.t.u. per hour, 110,000 pounds of air must, therefore, be recirculated, and the total amount passed through the air washers will be double this quantity, or 220,000 pounds per hour.

The average temperature of the air entering the washer will be $\frac{95 + 80}{2}$ or 87.5° F., and the average relative humidity may be found as follows:

One pound of air at 95° F. and 50 per cent relative humidity contains 123 grains of moisture, and 1 pound of air at 80° F. and 70 per cent relative humidity contains 107 grains of moisture. The average moisture per pound of air is then $\frac{123 + 107}{}$ or 115 grains; and the temperature of the mixture entering the air washer will be 87.5° F. and its relative humidity will be 59 per cent.

For calculating the size of the ventilating fan the following data are needed: The volume per pound of saturated air at 87.5° F. is 14.4 cubic feet; of dry air at 87.5° F. is 13.77 cubic feet; of water vapor per pound of saturated air at 87.5° F. is 0.63 cubic feet; of water vapor per pound of air at 87.5° F. and 59 per cent relative humidity is 0.372 cubic foot; of air at 87.5° F. and 59 per cent relative humidity is $13.77 + 0.372$ or 14.14 cubic feet.

The fans must, therefore, have a capacity of $220{,}000 \times 14.14/60$ or 51,800 cubic feet per minute.

Total cooling to be done by the air washer is 1,440,000 B.t.u. per hour. Since 1 ton of refrigeration is equal to 12 000 B.t.u. per hour, the cooling by the air washer will be 120 tons.

Each pound of air entering the theater must absorb 720,000/220,000 or 3.27 B.t.u. One pound of air at 80° F. and 70 per cent relative humidity contains 35.27 B.t.u., and 1 pound of air leaving the air washer contains 35.27 − 3.27 B.t.u. or 32.00 B.t.u. Therefore, the wet-bulb temperature leaving the washer must be 68° F.

Checking the calculations of Example 1, the tons of refrigeration required to cool the air entering the washer at 87.5° F. and 59 per cent relative humidity to a temperature of 68.5° F. wet-bulb temperature, the heat removed per pound of air at 87.5° F. and 59 per cent relative humidity is 38.75 B.t.u.; and the heat removed per pound at 68.5° F. and 100 per cent relative humidity is 32.00 B.t.u.

The heat to be removed per pound of air is, therefore, 6.75 B.t.u. and the refrigeration required is 220,000 × 6.75/12,000, or approximately 123 tons.

The fan capacity will, therefore, be 51,800 cubic feet per minute, the water in the air washer must be cooled to 68° F., and 120 tons of refrigeration will be required in the air washer.

Example 14: It is desired to furnish 50,000 cubic feet of air per minute at 75° F. and 70 per cent relative humidity to a moving-picture theater, when the outside temperature is 96° F. and the relative humidity is 35 per cent. What will be the tonnage of refrigeration required at the air washer, and to what temperature must the water in the air washer be cooled?

Air at 75° F. and 70 per cent relative humidity contains 90 grains per pound, and the dew point is 64.5° F. The amount of heat in a pound of air at 96° F. and 35 per cent relative humidity is 36.50 B.t.u., and the heat per pound of air at 64.5° F. and 100 per cent relative humidity is 29.00 B.t.u., making the heat to be absorbed per pound of air 7.50 B.t.u.

One pound of air at 96° F. and 100 per cent relative humidity contains 14.85 cubic feet, and 1 pound of dry air at 96° F. contains 14.00 cubic feet. The volume of vapor per pound of air is, therefore, 0.85 cubic foot.

The volume of vapor per pound of air at 96° F. and 35 per cent relative humidity is, therefore, 0.85 × 0.35 or 0.30 cubic foot, and the volume of 1 pound of air at 96° F. and 35 per cent relative humidity is 14.00 + 0.30 or 14.30 cubic feet. The total heat in B.t.u. to be absorbed per hour is therefore 50,000 × 60 × 7.5/14.30, and the tons of refrigeration required will be 50,000 × 60 × 7.5/14.30 × 12,000 or 131 tons. The water in the air washer must be cooled to a temperature corresponding to the dew point of air at 75° F. and 70 per cent relative humidity or 64.5° F. Where extremely low relative humidity is required it is necessary to cool the air to a much lower temperature in the air washer and to reheat the air before introducing it into the room.

Example 15: For a manufacturing process, 5,000 pounds of air per minute at 70° F. and 20 per cent relative humidity are to be provided, when the outside air is at 85° F. and the relative humidity is 45 per cent. What should be the temperature of the water in the air washing? How many tons of refrigeration will be required at the air washer? How much steam at 5 pounds per square inch gage pressure must be supplied per hour to the reheating coils?

Air at 70° F. and 20 per cent relative humidity contains 21 grains of moisture, and the dew point is at 27° F.; the heat per pound of air at 85° F.

and 45 per cent relative humidity is 32.5 B.t.u.; the heat per pound of air at 27° F. and 100 per cent relative humidity is 9.75 B.t.u.; and, therefore, the heat to be removed per pound of air is 22.75 B.t.u.; the refrigeration required being 5,000 × 60 × 22.75/12,000 or 570 tons.

The heat per pound of air at 70° F. and 20 per cent relative humidity is 20.25 B.t.u.; the heat per pound of air at 27° F. and 100 per cent relative humidity is 9.75 B.t.u., so that the heat to be supplied per pound of air is 10.5 B.t.u.

One pound of steam at 5 pounds per square inch gage pressure will give up, when condensing, 961 B.t.u., and the total pounds of steam required per hour will be, therefore, 10.50 × 5,000 × 60/961 or 3,300 pounds.

As the temperature required in the air washer is below freezing it will be necessary to use brine for cooling purposes in place of water.

The ventilation system for a theater should normally be designed to provide about eight *changes of air* per hour. A house seating 2,000 people normally requires about 50,000 cubic feet of air per minute. Good results are usually obtained by providing 2.5 tons of refrigeration per 1,000 cubic feet of fan capacity per minute. In southern cities it is usual to increase the capacity about 25 per cent. It is good practice to recirculate 50 per cent of the air, as this considerably decreases the refrigeration required and still provides ample fresh air for proper ventilation. The water-circulation pump for the air washer should have at least 50 per cent more capacity than the size of the pump ordinarily provided for an air washer on a ventilation system, to provide sufficient spray water for the direct expansion coils (p. 70). Normally about 3.5 gallons of water per minute per ton of refrigeration will be required. It will be necessary to provide about 35 lineal feet 1¼-inch direct-expansion piping in the air washer per ton of refrigeration. The spray chamber should be designed to give an air velocity of not over 500 feet per minute.

Refrigeration for Air Conditioning.—For conditioning air for ventilation when it is necessary to not only cool the air but to reduce its relative humidity, mechanical refrigeration is often required when an ample supply of cold water is not available. The problems of air conditioning will, therefore, frequently require for their solution a knowledge of the present practice in mechanical refrigeration as it applies to cases of this kind. The general types of refrigerating equipment for this service is, of course, the same as those already described. A diagrammatic representation of a refrigerating system as it is applied to the cooling of an air washer is shown in Fig. 288; the air washer serving in this place for cooling and dehumidifying the air supplied to a large building. Direct-expansion coils made of wrought iron or steel pipe are located in the air washer in such positions that the water sprayed from the nozzles in the washer will pass over them before falling into the tank at the bottom of

the washer. Each coil thus used for cooling is provided with an expansion valve to control the flow of the refrigerant.

Determination of Compression Pressures.—After the temperature to which the water in the air washer must be cooled has been determined (p. 463), the pressure required in the expansion coils to obtain that temperature can be found, knowing that it will be necessary to maintain in the coils a temperature about 30° F. lower than the temperature of the water that has been cooled in the washer. If, for example, the temperature in the coils is to be 25° F. and the refrigerant is ammonia, it will be found from the tables of the properties of this refrigerant that

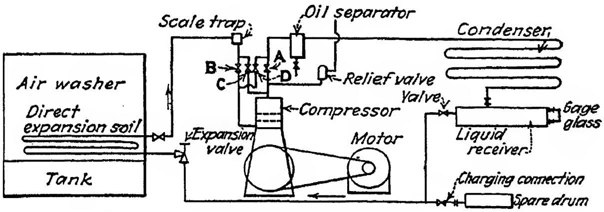

FIG. 288.—Refrigerating system applied to air washer.

the pressure corresponding to this temperature is approximately 39 pounds per square inch gage pressure; meaning, therefore, that the suction pressure of the compressor will have this value.

Carrier System of Air Conditioning.—The system of air conditioning developed by the Carrier Engineering Corporation uses a centrifugal compressor (p. 131); the refrigerant being dichloromethane or carrene (p. 89). The compressor, condenser, and cooler form an integral unit, while the condenser and cooler serve as a base for the compressor (p. 62). The liquid refrigerant in the cooler flows over a large number of bronze tubes through which the water is circulated to cool the spray chamber of the air washer. In the normal operation of the compressor, a vacuum of 25 inches of mercury is maintained in the expansion coils of the cooler, which is the boiling point of the refrigerant at a temperature of 35° F. The vapor of the refrigerant is discharged from the compressor into the condenser where it is to be condensed, there being a vacuum also in the condenser of 10 inches of mercury. In both the condenser and the cooler, the vapor of the refrigerant is on the outside of the coils, the cooling water in both cases flowing through the tubes. A thermo-electric control maintains automatically a uniform

temperature of the spray water in the cooler, the control device operating by starting and stopping a pump used to circulate the liquid refrigerant.

In cold-storage and ice-making plants where lower temperatures are required than for air conditioning, it is usual to adjust the valves on the expansion coils so that there will be a slight frost on the suction connections to the compressor, but with the relatively higher temperatures ordinarily used for air conditioning it is not desirable to carry a sufficiently low pressure to make

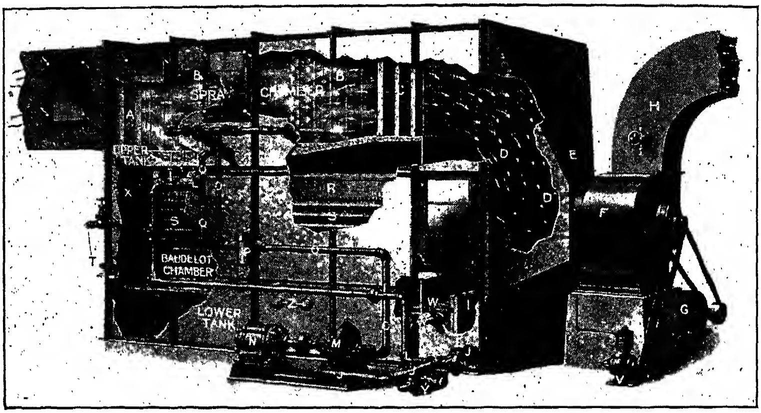

FIG. 289.—Air dehumidifier and cooler.

this frost, the reason being that the lower suction pressure requires the handling of a larger volume of vapor per ton of refrigeration than would otherwise be necessary.

Dehumidifier.—The following is a description of an air dehumidifier made by the Carrier Engineering Corporation and shown in Fig. 289. This dehumidifier is of the self-contained type and consists of nozzles for spraying recirculated water into air which enters through the distributor plates *A* and leaves at *D*, after passing through the dust- and germ-eliminating plates *C*. These sprays are located in the spray chamber *B*, below which is the trough *R* for distributing the water over the direct-expansion coils *S*, located in the *Baudelot chamber*. The spray nozzles are so arranged that they cause a uniformly dense bank of mist through which the air must pass on its way to the staggered dust- and germ-eliminating plates. The air is scrubbed

by the wet surfaces of the plates and freed from nearly all solid foreign matter, including disease germs.

The air after being washed enters the fan F, which is of the centrifugal type, passes through the fan-inlet connection E, and then out of the fan-outlet connection H into the duct system. The fan is driven by an electric motor G.

The water is taken from the bottom of the tank through the screen I into the centrifugal pump M, by way of the pipe line J. The centrifugal pump is driven by an electric motor N. The water which is discharged from the pump passes through the pipe line O and the pot strainer P, which is used to remove any

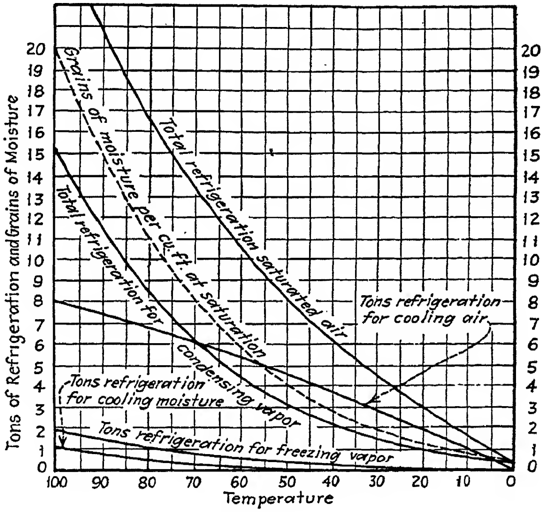

FIG. 290.—Refrigeration required for air conditioning.

scale and dirt which may be carried by the water supplied to the spray nozzles. A bypass Q connected to the "upper tank" is used for quickly cooling the water when first starting the apparatus. The pipes carrying the refrigerant (entering and leaving) are marked by the letter T in the figure.

An overflow W and a drain to the sewer Y are provided for the "lower tank," and there is also a drain X for the upper tank. The fresh-water connection Z is for supplying the make-up and cleaning water. A small air compressor V driven from the fan shaft supplies air to a thermostat which controls the temperature of the air leaving the apparatus. A three-way valve K in the

water line J has a pipe line L connected to it, which leads to the troughs R. This three-way valve, which is also connected into the suction line of the spray pump M, is used for regulating the amount of water taken from the settling tank of the dehumidifier and from the cold-water supply.

Humidostat.—For the control of the relative humidity an instrument called a "humidostat" is used. In place of the sensitive element provided in a temperature-controlling device (thermostat), a substance which is affected by the relative humidity of the atmosphere, and not by temperature, is used. The humidostat is made to be located on the wall of a room or inserted in an air duct. To secure the best results with humidity control, it is necessary that the temperature within the building be maintained constant with automatic temperature regulation; and where a system of temperature control is installed, the humidity control may be added at very little extra expense.

Table XVIII shows the temperature and humidity commonly used in the listed manufacturing processes. The cost of ventilation and air conditioning in buildings of various types is given in Table XIX.

A chart for determining the tonnage of refrigeration required for air-conditioning purposes per 1,000 cubic feet of air per minute is shown in Fig. 290. From this chart one can obtain the number of tons of refrigeration to cool the air, condense the water vapor, and cool and freeze the water vapor in the air. For example, the tons of refrigeration required to cool 1,000 cubic feet of saturated air at 70 to 0° F. are 13.25; from 50 to 0° F. are 8.2. Hence, to cool from 70 to 50° F. the refrigeration needed is 5.05 tons. The total refrigerating effect required is the sum of the following amounts (as shown by the curves): for cooling air 6.1, for condensing moisture 6.1, for freezing moisture 0.75, and for cooling moisture 0.3—a total of 13.25 tons. If the original air is 80 per cent saturated, then the refrigeration required to condense, cool, and freeze the moisture is less in proportion.

TABLE XVIII.—TEMPERATURE AND HUMIDITY
For Manufacturing Processes

Industry and product	Process	Temperature, degrees Fahrenheit	Relative humidity, per cent
Cotton	Carding	68 –73.4	50
	Combing	68 –73.4	60–65
	Roving	68 –73.4	50–60
	Spinning	68 –73.4	60–65
	Spooling, twisting	68 –73.4	65
	Warping	68 –73.4	65
	Weaving	68 –73.4	75–80
Wool	Carding	73.4–77	65–70
	Spinning	73.4–77	55–60
	Weaving	68 –73.4	50–55
	Storage for shipping	68 –73.4	55–60
Silk	Dressing	69.8–77	60–65
	Spinning	69.8–77	65–70
	Throwing	69.8–77	65–70
	Weaving	69.8–77	60–70
Confectionery	Chocolate enrobing	64.4	55
	Chocolate enrobing, hot end	80	30–35
	Hard candy making	69.8	50
	Storage	40–60	50–70
Tobacco	Softening	84.2	85
	Cigar and cigarette making	69.9–73.4	55–70
Printing	Lithographing	69.9	45
	Relief and offset	77.0	45
	Folding	77.0	65
	Binding	69.9	45
Baking	Dough fermentation	80.6	65–70
	Proofing	89.6–95	80–90
	Loaf cooling	69.9	65
Electrical cable	Winding insulation	104	5

TABLE XVIII.—TEMPERATURE AND HUMIDITY (*Continued*)

Industry and Product	Process	Temperature, degrees Fahrenheit	Relative humidity, per cent
Cellulose lacquers	Application	75.2	20
Munitions	Fuse loading	69.9	55
Cereals	Seal packing prepared crisp cereals	73.4	45–50
Fruits	Apple storage	31–34	80–85
	Avocado packing	40	50
	Bananas:		
	Holding ripe fruit	56	70–75
	Holding green fruit	58	70–75
	Slow ripening	60–62	90
	Normal ripening	64–68	90
	Fast ripening	70–72	90
	Danger of chilling	Below 34	
Dairy products	Butter manufacturing	60	60
	Chill room	40	60
Chewing gum	Rolling and scoring chicle	75	50
	Wrapping and packing	70	45
Prepared powdered beverages and crisp cereals		75	35–40
Sugar storage		80	35
Meat products	Butter substitutes:		
	Churn room	70	60
	Print room	60	60
	Chill room	30	60
	Cooler	55	60
	Bacon slicing	60	48

TABLE XIX.—AVERAGE COST OF VENTILATION AND AIR CONDITIONING*

Type of building	Initial cost per square foot of floor space	Annual over-all cost per square foot of floor space	Unit cost, based on annual over-all cost	Annual power costs, rate in parenthesis
Department store, 160,000 square feet of floor space, 6,250 people.	$1.80	0.32	32 cents per year per square foot of sales area	$15,923 (3 cents per kilowatt-hour)
Office building, 325,000 square feet of floor space, 2,540 people.	1.25	0.19	1 cent per man-hour	$13,896 (1.2 cents per kilowatt-hour)
Plant office building, 42,560 square feet of floor space, 300 people.	2.00	0.31	3 cents per man-hour	$2,694 (3 cents per kilowatt-hour)
Restaurant and cafeteria, 12,644 square feet, 760 people.	3.60†	1.16†	1 cent per meal	$4,748† (3 cents per kilowatt-hour)
Hotel (guest rooms), 120,000 square feet 600 people	1.50	0.50‡	20 cents per room per day	$13,600 (3 cents per kilowatt-hour)

Note that both the initial and annual costs are based on heating, ventilating, and conditioning. In most cases, heating and ventilating are normal expenses which must be paid anyway. The actual cost for adding air conditioning, which also provides for heating and ventilating, is on the average one-third of the figures used above.

* CARRIER, W. H., "Recent Progress in Air Conditioning," *Refrigerating Eng.*, Vol. 21, No. 3, pp. 188–189.

† Higher initial and operating costs are due to a larger required capacity to provide for unusual heat load, more frequent air change, and larger number of people.

‡ Higher operating cost is due to the longer period of the daily operation.

APPENDIX A

PROBLEMS IN REFRIGERATION

1. A creamery must be equipped to cool 1,000 pounds of milk from an initial temperature of 90 to a final temperature of 35° F. How much heat must be removed? Specific heat of milk is 0.95.

2. In problem 1, how many pounds of brine must be supplied to remove this heat if the allowable rise in temperature is 10° F. and the specific heat of the brine is 0.837?

3. How many heat units must be removed from 1 pound of ammonia vapor at atmospheric pressure and 80° F. to liquefy it?

4. A refrigerating room is held at 10° F. by a dense-air machine and operated under the conditions shown by Fig. 291. Determine (*a*) the temperature at point 4 in the figure; (*b*) heat removed by 1 pound of air; (*c*) weight of air per minute per ton of refrigeration; (*d*) net work per minute per ton of refrigeration, assuming friction loss to be 15 per cent; (*e*) weight

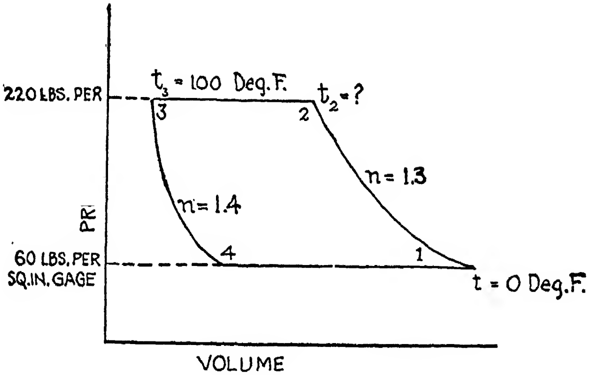

FIG. 291.—Cycle of dense-air compressor.

of cooling water per minute per ton of refrigeration, assuming temperature of entering cooling water to be 60° F. and temperature of outlet cooling water to be 70° F.; (*f*) horsepower required to drive machine per ton of refrigeration; (*g*) displacement per minute per ton of refrigeration for the compressor, assuming 2 per cent clearance; (*h*) displacement per minute per ton of refrigeration for expander, assuming 2 per cent clearance; (*i*) refrigerating effect or coefficient of performance.

5. A cold-storage room is held at 60° F. by an air-refrigerating machine which does not use the air over and over again. The cycle is made up of two adiabatic lines (exponent $n = 1.4$) and two constant-pressure lines. The suction pressure is 15 pounds per square inch absolute, and the discharge pressure is 100 pounds per square inch absolute. The temperature

of the air entering the air motor or expander is 70° F., and the inlet and outlet temperatures of the cooling water are, respectively, 65 and 90° F. Compressor and expander operate at 200 r.p.m. Find (*a*) the refrigeration per pound of air; (*b*) heat removed by the cooling water per pound of air; (*c*) pounds of cooling water required to cool 1 pound of air; (*d*) net work per pound of air; (*e*) displacement of compressor per revolution, assuming 2 per cent clearance; (*f*) displacement of air motor or expander per revolution, assuming 2 per cent clearance; (*g*) refrigerating effect or coefficient of performance.

6. How much work is done when 4 pounds of ammonia occupying 8 cubic feet at 141.7 pounds per square inch absolute pressure expands adiabatically to pressure corresponding to saturation temperature of 30° F.? Also, find the value of the exponent n in this case.

7. Find the amount of work done in compressing 1 pound of *dry* ammonia vapor at a pressure of 15.98 pounds per square inch absolute to the condition of 155 pounds per square inch absolute pressure and 200° F. superheat. Also, determine the value of the exponent n.

8. The suction pressure of an ammonia compressor is 30.57 pounds per square inch absolute, and the discharge pressure is 182 pounds per square inch absolute. If the ammonia vapor is compressed adiabatically, find the work done by one pound when the quality of the vapor at the end of compression is 100 per cent. Obtain the value of the exponent n, and check the work done by the use of the expression, $\frac{P_2V_2 - P_1V_1}{n - 1}$.

9. The temperature of liquid ammonia in the liquid receiver is 75° F., and the pressure in the evaporating coils is 35 pounds per square inch absolute. Find (*a*) the amount of heat used to cool 1 pound of liquid ammonia; (*b*) weight of ammonia evaporated to do this cooling; (*c*) net cooling effect of the pound of original ammonia.

10. The discharge gage pressure of a compressor is 185 pounds per square inch, and liquid ammonia is allowed to enter the expansion valve at 70° F. The evaporating-coil gage pressure is 5 pounds per square inch. Find (*a*) the amount of heat used to cool 1 pound of liquid ammonia; (*b*) weight of ammonia evaporated to do this cooling; (*c*) net cooling effect of the pound of the original ammonia; (*d*) quality of the ammonia, just after it has been cooled to the temperature in the evaporating coil.

11. Find the mean specific heat of 1 pound of liquid ammonia having initial and final temperatures of 70 and 10° F., respectively.

12. The suction pressure of an ammonia compressor is 38 pounds per square inch absolute, and the discharge pressure is 140 pounds per square inch absolute. How many pounds of ammonia per minute must be circulated to produce 1 ton of refrigeration?

13. An ammonia compressor circulates 45 pounds of ammonia per hour at a discharge gage pressure of 175 pounds per square inch. The condensing water enters at 70° F. and leaves at 80° F. The temperature of the liquid ammonia entering the expansion valve is 80° F. How many gallons of water must be supplied to the condenser per hour?

14. If, in problem 13, the liquid ammonia reaches the expansion valve at 75° F. and the temperature in the expansion coils is 15° F., find the number of heat units required to lower its temperature to the boiling point. How

many tons of refrigeration are produced by the cooling coils when this amount of ammonia is circulated?

15. How many pounds of carbon dioxide must be circulated per minute per ton of refrigeration if the pressure in the expansion coils is 284.6 pounds per square inch absolute and the carbon dioxide reaches the expansion valve at a temperature of 77° F.? For the properties of carbon dioxide, Appendix (Fig. 294).

16. How many pounds of ammonia are necessary to fill an expansion coil which has 12,000 feet of 2-inch pipe, if the gage pressure is 25 pounds per square inch. Assume the quality of the vapor in the suction line to be 95 per cent.

17. How many pounds of ammonia must be circulated per minute for a 50-ton plant if the suction-gage pressure is 25 pounds per square inch and the temperature of the liquid ammonia at the expansion valve is 80° F.?

18. A compressor of 100-ton rating operates with a discharge gage pressure of 165.3 pounds per square inch and a suction gage pressure of 15.3 pounds per square inch. If the vapor is superheated 25° F. when it enters the compressor, determine the following quantities, assuming the temperature of the liquid ammonia at the expansion valve to be 85° F.: (*a*) weight of ammonia per minute per ton of refrigeration; (*b*) horsepower required by compressor per ton of refrigeration; (*c*) piston displacement per minute per ton of refrigeration; (*d*) coefficient of performance.

19. A compressor operates with a discharge pressure of 180 pounds per square inch absolute and a suction pressure of 18 pounds per square inch absolute. If the vapor is superheated 40° F. when it enters the compressor, determine the following quantities, assuming the temperature of the liquid ammonia at the expansion valve to be 85° F.: (*a*) weight of ammonia per minute per ton of refrigeration; (*b*) horsepower required by compressor per ton of refrigeration; (*c*) piston displacement per minute per ton of refrigeration; (*d*) coefficient of performance.

20. Arrange the results of problems 18 and 19 in a table, with the headings showing the effect of the pressures and temperatures upon the performance of the machine.

21. An ammonia compressor operates with dry compression and discharges the vapor at 175 pounds per square inch gage pressure. If the suction-gage pressure is 23 pounds per square inch, what horsepower per ton of refrigeration is required to drive the compressor and engine, the overall efficiency being 80 per cent?

22. What size of double-acting ammonia compressor is necessary to produce 100 tons of refrigeration per 24 hours at a discharge-gage pressure of 120 pounds per square inch and a suction-gage pressure of 15 pounds per square inch? Assume a volumetric efficiency of 75 per cent and dry compression.

23. If ammonia vapor is compressed so that it is dry at the end of compression, determine the number of gallons of cooling water required per minute per ton of refrigeration if the condenser-gage pressure is 150 pounds per square inch and the suction-gage pressure is 25 pounds per square inch. Assume a 10° F. difference in temperature between the liquid ammonia and the leaving cooling water. The liquid ammonia leaves the condenser at 75° F., and the cooling water enters at 60° F.

24. A compressor discharges ammonia vapor to a condenser at a gage pressure of 190 pounds per square inch. The suction-gage pressure is 19 pounds per square inch, and the ammonia vapor is superheated 20° F. between the evaporating coils and the compressor. Find the temperature of the discharged ammonia vapor and the amount of heat removed (*a*) to extract the heat of the superheat, (*b*) to liquefy the ammonia vapor, and (*c*) to cool the liquid ammonia to 80° F.

25. In problem 24, what percentage of the total heat removed is the heat which is absorbed (*a*) to remove the superheat, (*b*) to liquefy the ammonia vapor, and (*c*) to cool the liquid ammonia? What is the useful refrigeration per pound of ammonia and the heat loss due to superheating the ammonia vapor in the suction line?

26. In problem 24, find the number of pounds of ammonia circulated per minute per ton of refrigeration. If an ice plant requires 1.6 tons of refrigeration to make 1 ton of ice, find the amount of ammonia to be circulated per minute per ton of ice.

27. In problem 24, find the volume of ammonia vapor passing through the compressor per minute per ton of refrigeration and per ton of ice. Assume a volumetric efficiency of 75 per cent. Determine the number of gallons of water per ton of refrigeration and per ton of ice made, assuming a rise in temperature of 10° F. of the cooling water.

28. In problem 24, determine the horsepower required to drive the compressor per ton of refrigeration and ton of ice. Assuming an efficiency of 85 per cent for motor and compressor, what size motor (horsepower) would be required for a 100-ton ice plant?

29. In problem 24, what size double-acting ammonia compressor is required to produce 100 tons of ice per day, and what is the speed at which it is to be driven?

30. A compressor is designed to operate with a temperature of 0° F. in the evaporator and a liquefaction temperature of 96° F., but is operated, instead, at a temperature of 5° F. in the evaporator and at 86° F. liquefaction temperature. Find the increase in capacity over its normal rating. What effects have the increase of evaporator pressure and the decrease of liquefaction temperature upon the capacity of the compressor?

31. The evaporating coils in a refrigerating plant are held at a temperature of 5° F., and the liquid ammonia enters the expansion valve at 80° F. If the liquid ammonia is cooled to 60 instead of 80° F., what is the percentage gain in refrigerating effect by this aftercooling? How many degrees (Fahrenheit) of aftercooling are necessary to gain 1 per cent in refrigerating effect?

32. A compressor operates at a capacity of 100 tons of refrigeration with a discharge-gage pressure of 185 pounds per square inch and a discharge temperature of 263° F. If the suction-gage pressure is 10 pounds per square inch and the vapor is *dry* at the suction valve of the compressor, find the size of discharge and suction pipe if the velocities are 8,000 and 4,000 feet per minute, respectively.

33. In the above problem, if the distance between the compressor and the condenser is 120 feet, find the condenser pressure, assuming that the pressure drop ($P_1 - P_2$) is expressed in pounds per square inch by the following equation:

$$P_1 - P_2 = \frac{V^2L(1 + 3.6 \div d)D}{144 \times 454 \times d}$$

where

V = velocity of the ammonia vapor, feet per second
L = length of pipe, feet
D = density of ammonia vapor at the pressure P_1, pounds per cubic foot
d = diameter of pipe, inches

34. The following data were obtained from a test of a double-acting compressor of 15 tons capacity which was driven by a steam engine.

Size of steam cylinder	9 × 24 in.
Diameter of piston rod	2 in.
Size of compressor cylinder	8 × 16 in.
Diameter of piston rod	2 in.
Duration of test, hours	1.5
Suction pressure, pounds per square inch gage	15
Condenser pressure, pounds per square inch gage	120
Revolutions per minute	72
Temperature of brine, inlet, degrees Fahrenheit	36.4
Temperature of brine, outlet, degrees Fahrenheit	11.1
Difference of inlet and outlet temperatures, degrees Fahrenheit	
Specific heat of brine	0.757
Weight of brine circulated, pounds	8,025
Weight of brine circulated per hour, pounds	
Refrigeration produced, B.t.u. per hour	
Capacity developed, tons per 24 hours	
Temperature of outlet condensing water, degrees Fahrenheit	68
Temperature of inlet condensing water, degrees Fahrenheit	55
Difference of temperature of outlet and inlet of condensing water, degrees Fahrenheit	
Weight of cooling water used, pounds	19,730
Weight of cooling water used per hour, pounds	
B.t.u. absorbed per hour by cooling water	
Ammonia temperatures, inlet to condenser, degrees Fahrenheit	74.7
Ammonia temperatures, outlet to condenser, degrees Fahrenheit	56.7
Ammonia temperature, difference of outlet and inlet, degrees Fahrenheit	
Ammonia temperature at cooler, inlet, degrees Fahrenheit	64.8
Ammonia temperature at cooler, outlet, degrees Fahrenheit	0.7
Ammonia temperature difference, cooler inlet and outlet, degrees Fahrenheit	
Weight of ammonia circulated, pounds	427.5
Weight of ammonia circulated, per hour	
Weight of dry steam used, pounds	1,049
Weight of dry steam used, per hour	
Mean effective pressure, head end, steam cylinder, pounds per square inch	34.9
Mean effective pressure, crank end, steam cylinder, pounds per square inch	30.45

Mean effective pressure, head end, ammonia cylinder, pounds per square inch	49.7
Mean effective pressure, crank end, ammonia cylinder, pounds per square inch	50.8
Indicated horsepower, head end, steam cylinder	
Indicated horsepower, crank end, steam cylinder	
Indicated horsepower, total, steam cylinder	
Indicated horsepower, head end, ammonia cylinder	
Indicated horsepower, crank end, ammonia cylinder	
Indicated horsepower, total, ammonia cylinder	
Mechanical efficiency, per cent	
Weight of dry steam per indicated horsepower per hour, steam cylinder, pounds	
Weight of dry steam per hour per ton of refrigeration, pounds	

HEAT BALANCE

	Heat gained B.t.u.	Heat lost B.t.u.
Work of compression		
Between compressor and condenser		
To condensing water		
Between condensers and cooler		
In cooler		
Total		

35. What temperature is required to make a 40 per cent solution of aqua ammonia boil at a pressure of 120 pounds per square inch absolute?

36. A 30 per cent solution of aqua ammonia boils in a generator at 200° F. What is the pressure in the generator?

37. A 36 per cent solution of aqua ammonia enters a generator and boils at 214° F. If the generator gage pressure is 122 pounds per square inch, find the weight of weak aqua ammonia required to absorb 1 pound of ammonia vapor.

38. An absorber operates at a gage pressure of 10 pounds per square inch. If the strong aqua ammonia leaves the absorber at 80° F., what is the strength of the aqua ammonia?

39. How much heat is produced when 5 pounds of ammonia vapor are absorbed in 45 pounds of water? How much heat is required to drive off this same amount of ammonia from the solution which has been made?

40. How much heat will be generated in an absorber and required in a generator per pound of ammonia vapor for the following conditions: strong aqua ammonia, 33 per cent; weak aqua ammonia, 22 per cent?

41. A 35 per cent solution of aqua ammonia leaves an absorber, and a 28 per cent solution of aqua ammonia enters an absorber, how much heat will be liberated per minute when 40 pounds of ammonia vapor are absorbed per minute?

42. A generator operates at a gage pressure of 122 pounds per square inch. If the ammonia vapor leaving the generator is at a temperature of 208° F., find the partial steam pressure in the generator. What is the ammonia *vapor pressure* (see p. 96)?

43. How many pounds of strong aqua ammonia must be circulated per pound of ammonia if the concentrations are as follows: strong aqua ammonia, 35 per cent; weak aqua ammonia, 27 per cent?

44. With the conditions in the above problem, and the evaporating coils held at 0° F., how many pounds of strong aqua ammonia must be circulated per minute per ton of refrigeration if the liquid ammonia enters the expansion valve at 80° F.?

45. What is the heat of the liquid of a 20 per cent solution of water and ammonia at 180° F.?

46. Strong aqua ammonia of 35 per cent concentration enters a generator at 150° F., and weak aqua ammonia of 25 per cent concentration leaves the generator at 200° F.; how much heat must be added to the aqua ammonia per pound of ammonia vaporized?

47. A testing box passes 7.9 B.t.u. per hour *through a sample of insulation.* If the inside temperature is 70.6° F. and the outside temperature is 40.7° F., what is the value of the heat-transfer coefficient K if the area of the sample is 3.06 square feet?

48. If the temperatures of the surfaces of the sample in the above problem are 69.1° F. on the inside and 43.5° F. on the outside, what are the surface coefficients and the constant of conduction if the sample is 3 inches thick?

49. One side of a testing box has an outside area of 3.92 square feet and an inside area of 2.2 square feet. If the temperature difference is 25.7° F. and the heat passing through the material is 7.92 B.t.u. per hour, determine the coefficient of conductivity per square foot per 24 hours per degree Fahrenheit per inch of thickness. The test specimen is 3 inches thick.

50. The heat lost per hour through a wall 10 by 10 feet is 800 B.t.u. If the temperatures of the outside air is 90° F. and the inside air is 10° F., find the total loss per square foot per hour per degree Fahrenheit.

51. In a double-pipe condenser, the water velocity is 2.5 feet per second. Determine the number of linear feet of 1¼-inch pipe required for a 40-ton refrigerating plant operating at a compressor discharge pressure of 200 pounds per square inch absolute and a back pressure of 30 pounds per square inch absolute. The compression is dry, and the temperature of the liquid ammonia at the receiver is 85° F. The temperature rise of the cooling water is 10° F. and assume that the heat transmission coefficient is 200 B.t.u. per sq. ft. per hour per °F. difference.

52. A wall is constructed of 2-inch concrete on the outside, 12 inches of brick, a 1-inch air space, and 1 inch of plaster on the inside. Determine the total heat loss per hour per square foot per degree Fahrenheit.

53. A cold-storage room has an outside wall made of 10 inches of concrete, two courses of 2-inch cork board, and the inner surface covered with ½ inch of cement. Determine the coefficient of heat transmission for the wall with a 2-inch air space between the concrete and the cork board. Calculate, also, the coefficient of heat transmission without the air space between the concrete and the cork board.

54. How many tons of refrigeration are required to freeze 5,000 pounds of poultry from an initial temperature of 70° F. to a final temperature of 20° F.?

55. How much heat must be removed to lower the temperature of 1,000 tubs of butter (55 pounds each) from 60 to 10° F.?

56. An ice-storage house has a capacity of 2,000 tons of ice. The house has a floor area of 45 by 50 feet and is 45 feet high. The walls and ceilings are insulated with two layers of 2-inch cork board. The room is held at 22° F. when the outside temperature is 100° F. If the leakage loss for doors, lights, and workmen is taken at 15 per cent of the insulation loss, determine the number of linear feet of 2-inch pipe required. Assume a transmission coefficient of 2 B.t.u. per 24 hours per degree Fahrenheit per square foot for the total thickness.

57. A creamery receives 3,000 gallons of milk a day, cooled from 75 to 38° F. in 3 hours. What is the capacity of a machine which will produce this cooling. Neglect all losses.

58. Suppose a small machine were used in the above creamery, operating 9 hours per day for cooling brine-storage tanks. How many tons capacity in 24 hours should this machine have?

59. How many gallons of calcium brine should be used in the storage tanks in problem 58 if the brine had a strength of 90° F. salinometer? The brine is warmed 20° F. in cooling the milk. Assume that the machine is run for 5 hours while cooling it.

60. A refrigerator is 5 feet high, 4 feet wide, and 3 feet deep. The heat-transfer coefficient is 0.2. How many pounds of ice melt per hour if the temperature of the refrigerator is 48° F. and the temperature of the outside air is 95° F.

61. A cold-storage compartment is 30 feet long, 20 feet wide, and 10 feet high. The heat-transfer coefficient is 0.094. The inside temperature is 28° F. How many tons of refrigerating capacity will be required to maintain this temperature if the temperature of the outside air is 80° F.?

62. A cold-storage compartment is 40 feet long, 30 feet wide, and 10 feet high. Both the end walls and one side wall are exposed to the outside temperature, which is 85° F. The other side wall adjoins another compartment kept at the same temperature. Each end wall contains one double window, and the side wall contains four double windows with air spaces. The temperature of the cold-storage compartments is maintained at 36° F. The construction of the walls of the compartment is such that the heat-transfer coefficient is 0.0785. The rooms above and below this compartment are at a temperature of 36° F. In this compartment are placed 70 tons of beef at a temperature of 90° F. This beef is removed at the end of 48 hours, and a new lot put in. The specific heat of beef is 0.68. How many tons of refrigerating capacity are required to keep this room and its contents cooled?

63. A cold-storage compartment 30 feet long by 25 feet wide by 10 feet high passes 4,000 heat units *per hour* through the walls. The temperature of this compartment is 36° F. The goods stored here *each day* require the removal of 400,000 B.t.u. The compartment is cooled by brine coils carrying brine at 25° F. How many running feet of 1½-inch pipe should the cooling coils contain? Assume that each square foot of coil surface will transfer 2.5 B.t.u. per hour for each degree difference in temperature.

64. Suppose that the amount of heat removed from the above cold-storage compartment is not known. How many running feet of 1½-inch pipe would be required for *direct-expansion* coils?

65. In problem 64, allow a drop in temperature of 6° F. between the ammonia in the expansion coils and the brine. What strength of salt brine would be used?

66. A certain refrigerating system requires 5,000 gallons of calcium brine. It is to be of such strength as to have a freezing point of $-1.40°$ F. If calcium chloride costs \$2.15 per 100 pounds, what will be the cost of making the brine?

NOTE.—Water weighs $8\frac{1}{3}$ pounds per gallon. Use specific-gravity values.

67. A refrigerating system circulates 30,000 pounds of calcium brine per hour to maintain the desired temperature. The brine has a strength of 92° on the salinometer. Later, this brine had its strength reduced to 68°, because it was too strong to maintain the desired temperature. If the brine is weakened to 68° salinometer, how much brine would have to be circulated per hour to maintain the same temperature? Assume that the required amount of refrigeration is the same in both cases.

APPENDIX B

TABLES AND CHARTS

Table I.—Saturated Ammonia: Temperature Table

Temperature, degrees Fahrenheit	Pressure		Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content		Latent heat, B.t.u./lb.	Entropy		Temperature, degrees Fahrenheit
	Absolute, lbs./in.²	Gage, lbs./in.²			Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
t	p	$g. p.$	V	$1/V$	h	H	L	s	S	t
−60	5.55	*18.6	44.73	0.02235	−21.2	589.6	610.8	−0.0517	1.4769	**−60**
−59	5.74	*18.2	43.37	0.02306	−20.[illegible]	590.0	610.1	−0.0490	1.4741	−59
−58	5.93	*17.8	42.05	0.02378	−19.[illegible]	590.4	609.5	−0.0464	1.4713	−58
−57	6.13	*17.4	40.79	0.02452	−18.[illegible]	590.8	608.8	−0.0438	1.4686	−57
−56	6.33	*17.0	39.56	0.02528	−17.0	591.2	608.2	−0.0412	1.4658	−56
−55	6.54	*16.6	38.38	0.02605	−15.9	591.6	607.5	−0.0386	1.4631	**−55**
−54	6.75	*16.2	37.24	0.02685	−14.8	592.1	606.9	0.0360	1.4604	−54
−53	6.97	*15.7	36.15	0.02766	−13.8	592.4	606.2	−0.0334	1.4577	−53
−52	7.20	*15.3	35.09	0.02850	−12.7	592.9	605.6	−0.0307	1.4551	−52
−51	7.43	*14.8	34.06	0.02936	−11.7	593.2	604.9	−0.0281	1.4524	−51
−50	7.67	*14.3	33.08	0.03023	−10.6	593.7	604.3	−0.0256	1.4497	**−50**
−49	7.91	*13.8	32.12	0.03113	− 9.6	594.0	603.6	−0.0230	1.4471	−49
−48	8.16	*13.3	31.20	0.03205	− 8.5	594.4	602.9	−0.0204	1.4445	−48
−47	8.42	*12.8	30.31	0.03299	− 7.4	594.9	602.3	−0.0179	1.4419	−47
−46	8.68	*12.2	29.45	0.03395	− 6.4	595.2	601.6	−0.0153	1.4393	−46
−45	8.95	*11.7	28.62	0.03494	− 5.3	595.6	600.9	−0.0127	1.4368	**−45**
−44	9.23	*11.1	27.82	0.03595	− 4.3	596.0	600.3	−0.0102	1.4342	−44
−43	9.51	*10.6	27.04	0.03698	− 3.2	596.4	599.6	−0.0076	1.4317	−43
−42	9.81	*10.0	26.29	0.03804	− 2.1	596.8	508.9	−0.0051	1.4292	−42
−41	10.10	*9.3	25.56	0.03912	− 1.1	597.2	598.3	−0.0025	1.4267	−41
−40	10.41	*8.7	24.86	0.04022	0.0	597.6	597.6	0.0000	1.4242	**−40**
−39	10.72	*8.1	24.18	0.04135	1.1	598.0	596.9	0.0025	1.4217	−39
−38	11.04	*7.4	23.53	0.04251	2.1	598.3	596.2	0.0051	1.4193	−38
−37	11.37	*6.8	22.89	0.04369	3.2	598.7	595.5	0.0076	1.4169	−37
−36	11.71	*6.1	22.27	0.04489	4.3	599.1	594.8	0.0101	1.4144	−36
−35	12.05	*5.4	21.68	0.04613	5.3	599.5	594.2	0.0126	1.4120	**−35**
−34	12.41	*4.7	21.10	0.04739	6.4	599.9	593.5	0.0151	1.4096	−34
−33	12.77	*3.9	20.54	0.04868	7.4	600.2	592.8	0.0176	1.4072	−33
−32	13.14	*3.2	20.00	0.04999	8.5	600.6	592.1	0.0201	1.4048	−32
−31	13.52	*2.4	19.48	0.05134	9.6	601.0	591.4	0.0226	1.4025	−31
−30	13.90	*1.6	18.97	0.05271	10.7	601.4	590.7	0.0250	1.4001	**−30**
−29	14.30	*0.8	18.48	0.05411	11.7	601.7	590.0	0.0275	1.3978	−29
−28	14.71	0.0	18.00	0.05555	12.8	602.1	589.3	0.0300	1.3955	−28
−27	15.12	0.4	17.54	0.05701	13.9	602.5	588.6	0.0325	1.3932	−27
−26	15.55	0.8	17.09	0.05850	14.9	602.8	587.9	0.0350	1.3909	−26
−25	15.98	1.3	16.66	0.06003	16.0	603.2	587.2	0.0374	1.3886	**−25**
−24	16.42	1.7	16.24	0.06158	17.1	603.6	586.5	0.0399	1.3863	−24
−23	16.88	2.2	15.83	0.06317	18.1	603.9	585.8	0.0423	1.3840	−23
−22	17.34	2.6	15.43	0.06479	19.2	604.3	585.1	0.0448	1.3818	−22
−21	17.81	3.1	15.05	0.06644	20.3	604.6	584.3	0.0472	1.3796	−21
−20	18.30	3.6	14.68	0.06813	21.4	605.0	583.6	0.0497	1.3774	**−20**
−19	18.79	4.1	14.32	0.06985	22.4	605.3	582.9	0.0521	1.3752	−19
−18	19.30	4.6	13.97	0.07161	23.5	605.7	582.2	0.0545	1.3729	−18
−17	19.81	5.1	13.62	0.07340	24.6	606.1	581.5	0.0570	1.3708	−17
−16	20.34	5.6	13.29	0.07522	25.6	606.4	580.8	0.0594	1.3686	−16
−15	20.88	6.2	12.97	0.07709	26.7	606.7	580.0	0.0618	1.3664	**−15**
−14	21.43	6.7	12.66	0.07898	27.8	607.1	579.3	0.0642	1.3643	−14
−13	21.99	7.3	12.36	0.08092	28.9	607.5	578.6	0.0666	1.3621	−13
−12	22.56	7.9	12.06	0.08289	30.0	607.8	577.8	0.0690	1.3600	−12
−11	23.15	8.5	11.78	0.08490	31.0	608.1	577.1	0.0714	1.3579	−11

* Inches of mercury below 1 standard atmosphere (29.92 in.).

TABLE I.—SATURATED AMMONIA: TEMPERATURE TABLE (*Continued*)

Temperature, degrees Fahrenheit	Pressure		Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content		Latent heat, B.t.u./lb.	Entropy		Temperature, degrees Fahrenheit
	Absolute, lbs./in.²	Gage, lbs./in.²			Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
t	p	$g.\ p.$	V	$1/V$	h	H	L	s	S	t
−10	23.74	9.0	11.50	0.08695	32.1	608.5	576.4	0.0738	1.3558	**−10**
− 9	24.35	9.7	11.23	0.08904	33.2	608.8	575.6	0.0762	1.3537	− 9
− 8	24.97	10.3	10.97	0.09117	34.3	609.2	574.9	0.0786	1.3516	− 8
− 7	25.61	10.9	10.71	0.09334	35.4	609.5	574.1	0.0809	1.3495	− 7
− 6	26.26	11.6	10.47	0.09555	36.4	609.8	573.4	0.0833	1.3474	− 6
− 5	26.92	12.2	10.23	0.09780	37.5	610.1	572.6	0.0857	1.3454	**− 5**
− 4	27.59	12.9	9.991	0.1001	38.6	610.5	571.9	0.0880	1.3433	− 4
− 3	28.28	13.6	9.763	0.1024	39.7	610.8	571.1	0.0904	1.3413	− 3
− 2	28.98	14.3	9.541	0.1048	40.7	611.1	570.4	0.0928	1.3393	− 2
− 1	29.69	15.0	9.326	0.1072	41.8	611.4	569.6	0.0951	1.3372	− 1
0	30.42	15.7	9.116	0.1097	42.9	611.8	568.9	0.0975	1.3352	**0**
1	31.16	16.5	8.912	0.1122	44.0	612.1	568.1	0.0998	1.3332	1
2	31.92	17.2	8.714	0.1148	45.1	612.4	567.3	0.1022	1.3312	2
3	32.69	18.0	8.521	0.1174	46.2	612.7	566.5	0.1045	1.3292	3
4	33.47	18.8	8.333	0.1200	47.2	613.0	565.8	0.1069	1.3273	4
5	34.27	19.6	8.150	0.1227	48.3	613.3	565.0	0.1092	1.3253	**5**
6	35.09	20.4	7.971	0.1254	49.4	613.6	564.2	0.1115	1.3234	6
7	35.92	21.2	7.798	0.1282	50.5	613.9	563.4	0.1138	1.3214	7
8	36.77	22.1	7.629	0.1311	51.6	614.3	562.7	0.1162	1.3195	8
9	37.63	22.9	7.464	0.1340	52.7	614.6	561.9	0.1185	1.3176	9
10	38.51	23.8	7.304	0.1369	53.8	614.9	561.1	0.1208	1.3157	**10**
11	39.40	24.7	7.148	0.1399	54.9	615.2	560.3	0.1231	1.3137	11
12	40.31	25.6	6.996	0.1429	56.0	615.5	559.5	0.1254	1.3118	12
13	41.24	26.5	6.847	0.1460	57.1	615.8	558.7	0.1277	1.3099	13
14	42.18	27.5	6.703	0.1492	58.2	616.1	557.9	0.1300	1.3081	14
15	43.14	28.4	6.562	0.1524	59.2	616.3	557.1	0.1323	1.3062	**15**
16	44.12	29.4	6.425	0.1556	60.3	616.6	556.3	0.1346	1.3043	16
17	45.12	30.4	6.291	0.1590	61.4	616.9	555.5	0.1369	1.3025	17
18	46.13	31.4	6.161	0.1623	62.5	617.2	554.7	0.1392	1.3006	18
19	47.16	32.5	6.034	0.1657	63.6	617.5	553.9	0.1415	1.2988	19
20	48.21	33.5	5.910	0.1692	64.7	617.8	553.1	0.1437	1.2969	**20**
21	49.28	34.6	5.789	0.1728	65.8	618.0	552.2	0.1460	1.2951	21
22	50.36	35.7	5.671	0.1763	66.9	618.3	551.4	0.1483	1.2933	22
23	51.47	36.8	5.556	0.1800	68.0	618.6	550.6	0.1505	1.2915	23
24	52.59	37.9	5.443	0.1837	69.1	618.9	549.8	0.1528	1.2897	24
25	53.73	39.0	5.334	0.1875	70.2	619.1	548.9	0.1551	1.2879	**25**
26	54.90	40.2	5.227	0.1913	71.3	619.4	548.1	0.1573	1.2861	26
27	56.08	41.4	5.123	0.1952	72.4	619.7	547.3	0.1596	1.2843	27
28	57.28	42.6	5.021	0.1992	73.5	619.9	546.4	0.1618	1.2825	28
29	58.50	43.8	4.922	0.2032	74.6	620.2	545.6	0.1641	1.2808	29
30	59.74	45.0	4.825	0.2073	75.7	620.5	544.8	0.1663	1.2790	**30**
31	61.00	46.3	4.730	0.2114	76.8	620.7	543.9	0.1686	1.2773	31
32	62.29	47.6	4.637	0.2156	77.9	621.0	543.1	0.1708	1.2755	32
33	63.59	48.9	4.547	0.2199	79.0	621.2	542.2	0.1730	1.2738	33
34	64.91	50.2	4.459	0.2243	80.1	621.5	541.4	0.1753	1.2721	34
35	66.26	51.6	4.373	0.2287	81.2	621.7	540.5	0.1775	1.2704	**35**
36	67.63	52.9	4.289	0.2332	82.3	622.0	539.7	0.1797	1.2686	36
37	69.02	54.3	4.207	0.2377	83.4	622.2	538.8	0.1819	1.2669	37
38	70.43	55.7	4.126	0.2423	84.6	622.5	537.9	0.1841	1.2652	38
39	71.87	57.2	4.048	0.2470	85.7	622.7	537.0	0.1863	1.2635	39
40	73.32	58.6	3.971	0.2518	86.8	623.0	536.2	0.1885	1.2618	**40**
41	74.80	60.1	3.897	0.2566	87.9	623.2	535.3	0.1908	1.2602	41
42	76.31	61.6	3.823	0.2616	89.0	623.4	534.4	0.1930	1.2585	42
43	77.83	63.1	3.752	0.2665	90.1	623.7	533.6	0.1952	1.2568	43
44	79.38	64.7	3.682	0.2716	91.2	623.9	532.7	0.1974	1.2552	44

TABLE I.—SATURATED AMMONIA: TEMPERATURE TABLE (*Continued*)

Temperature, degrees Fahrenheit	Pressure		Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content		Latent heat, B.t.u./lb.	Entropy		Temperature, degrees Fahrenheit
	Absolute, lbs./in.²	Gage, lbs./in.²			Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
t	p	$g.p.$	V	$1/V$	h	H	L	s	S	t
45	80.96	66.3	3.614	0.2767	92.3	624.1	531.8	0.1996	1.2535	**45**
46	82.55	67.9	3.547	0.2819	93.5	624.4	530.9	0.2018	1.2519	46
47	84.18	69.5	3.481	0.2872	94.6	624.6	530.0	0.2040	1.2502	47
48	85.82	71.1	3.418	0.2926	95.7	624.8	529.1	0.2062	1.2486	48
49	87.49	72.8	3.355	0.2981	96.8	625.0	528.2	0.2083	1.2469	49
50	89.19	74.5	3.294	0.3036	97.9	625.2	527.3	0.2105	1.2453	**50**
51	90.91	76.2	3.234	0.3092	99.1	625.5	526.4	0.2127	1.2437	51
52	92.66	78.0	3.176	0.3149	100.2	625.7	525.5	0.2149	1.2421	52
53	94.43	79.7	3.119	0.3207	101.3	625.9	524.6	0.2171	1.2405	53
54	96.23	81.5	3.063	0.3265	102.4	626.1	523.7	0.2192	1.2389	54
55	98.06	83.4	3.008	0.3325	103.5	626.3	522.8	0.2214	1.2373	**55**
56	99.91	85.2	2.954	0.3385	104.7	626.5	521.8	0.2236	1.2357	56
57	101.8	87.1	2.902	0.3446	105.8	626.7	520.9	0.2257	1.2341	57
58	103.7	89.0	2.851	0.3508	106.9	626.9	520.0	0.2279	1.2325	58
59	105.6	90.9	2.800	0.3571	108.1	627.1	519.0	0.2301	1.2310	59
60	107.6	92.9	2.751	0.3635	109.2	627.3	518.1	0.2322	1.2294	**60**
61	109.6	94.9	2.703	0.3700	110.3	627.5	517.2	0.2344	1.2278	61
62	111.6	96.9	2.656	0.3765	111.5	627.7	516.2	0.2365	1.2262	62
63	113.6	98.9	2.610	0.3832	112.6	627.9	515.3	0.2387	1.2247	63
64	115.7	101.0	2.565	0.3899	113.7	628.0	514.3	0.2408	1.2231	64
65	117.8	103.1	2.520	0.3968	114.8	628.2	513.4	0.2430	1.2216	**65**
66	120.0	105.3	2.477	0.4037	116.0	628.4	512.4	0.2451	1.2201	66
67	122.1	107.4	2.435	0.4108	117.1	628.6	511.5	0.2473	1.2186	67
68	124.3	109.6	2.393	0.4179	118.3	628.8	510.5	0.2494	1.2170	68
69	126.5	111.8	2.352	0.4251	119.4	628.9	509.5	0.2515	1.2155	69
70	128.8	114.1	2.312	0.4325	120.5	629.1	508.6	0.2537	1.2140	**70**
71	131.1	116.4	2.273	0.4399	121.7	629.3	507.6	0.2558	1.2125	71
72	133.4	118.7	2.235	0.4474	122.8	629.4	506.6	0.2579	1.2110	72
73	135.7	121.0	2.197	0.4551	124.0	629.6	505.6	0.2601	1.2095	73
74	138.1	123.4	2.161	0.4628	125.1	629.8	504.7	0.2622	1.2080	74
75	140.5	125.8	2.125	0.4707	126.2	629.9	503.7	0.2643	1.2065	**75**
76	143.0	128.3	2.089	0.4786	127.4	630.1	502.7	0.2664	1.2050	76
77	145.4	130.7	2.055	0.4867	128.5	630.2	501.7	0.2685	1.2035	77
78	147.9	133.2	2.021	0.4949	129.7	630.4	500.7	0.2706	1.2020	78
79	150.5	135.8	1.988	0.5031	130.8	630.5	499.7	0.2728	1.2006	79
80	153.0	138.3	1.955	0.5115	132.0	630.7	498.7	0.2749	1.1991	**80**
81	155.6	140.9	1.923	0.5200	133.1	630.8	497.7	0.2769	1.1976	81
82	158.3	143.6	1.892	0.5287	134.3	631.0	496.7	0.2791	1.1962	82
83	161.0	146.3	1.861	0.5374	135.4	631.1	495.7	0.2812	1.1947	83
84	163.7	149.0	1.831	0.5462	136.6	631.3	494.7	0.2833	1.1933	84
85	166.4	151.7	1.801	0.5552	137.8	631.4	493.6	0.2854	1.1918	**85**
86	169.2	154.5	1.772	0.5643	138.9	631.5	492.6	0.2875	1.1904	86
87	172.0	157.3	1.744	0.5735	140.1	631.7	491.6	0.2895	1.1889	87
88	174.8	160.1	1.716	0.5828	141.2	631.8	490.6	0.2917	1.1875	88
89	177.7	163.0	1.688	0.5923	142.4	631.9	489.5	0.2937	1.1860	89
90	180.6	165.9	1.661	0.6019	143.5	632.0	488.5	0.2958	1.1846	**90**
91	183.6	168.9	1.635	0.6116	144.7	632.1	487.4	0.2979	1.1832	91
92	186.6	171.9	1.609	0.6214	145.8	632.2	486.4	0.3000	1.1818	92
93	189.6	174.9	1.584	0.6314	147.0	632.3	485.3	0.3021	1.1804	93
94	192.7	178.0	1.559	0.6415	148.2	632.5	484.3	0.3041	1.1789	94
95	195.8	181.1	1.534	0.6517	149.4	632.6	483.2	0.3062	1.1775	**95**
96	198.9	184.2	1.510	0.6620	150.5	632.6	482.1	0.3083	1.1761	96
97	202.1	187.4	1.487	0.6725	151.7	632.8	481.1	0.3104	1.1747	97
98	205.3	190.6	1.464	0.6832	152.9	632.9	480.0	0.3125	1.1733	98
99	208.6	193.9	1.441	0.6939	154.0	632.9	478.9	0.3145	1.1719	**99**

TABLE I.—SATURATED AMMONIA: TEMPERATURE TABLE (*Continued*)

Temperature, degrees Fahrenheit	Pressure		Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content		Latent heat, B.t.u./lb.	Entropy		Temperature, degrees Fahrenheit
	Absolute, lbs./in.²	Gage, lbs./in.²			Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
t	p	$g.\ p.$	V	$1/V$	h	H	L	s	S	t
100	211.9	197.2	1.419	0.7048	155.2	633.0	477.8	0.3166	1.1705	**100**
101	215.2	200.5	1.397	0.7159	156.4	633.1	476.7	0.3187	1.1691	101
102	218.6	203.9	1.375	0.7270	157.6	633.2	475.6	0.3207	1.1677	102
103	222.0	207.3	1.354	0.7384	158.7	633.3	474.6	0.3228	1.1663	103
104	225.4	210.7	1.334	0.7498	159.9	633.4	473.5	0.3248	1.1649	104
105	228.9	214.2	1.313	0.7615	161.1	633.4	472.3	0.3269	1.1635	**105**
106	232.5	217.8	1.293	0.7732	162.3	633.5	471.2	0.3289	1.1621	106
107	236.0	221.3	1.274	0.7852	163.5	633.6	470.1	0.3310	1.1607	107
108	239.7	225.0	1.254	0.7972	164.6	633.6	469.0	0.3330	1.1593	108
109	243.3	228.6	1.235	0.8095	165.8	633.7	467.9	0.3351	1.1580	109
110	247.0	232.3	1.217	0.8219	167.0	633.7	466.7	0.3372	1.1566	**110**
111	250.8	236.1	1.198	0.8344	168.2	633.8	465.6	0.3392	1.1552	111
112	254.5	239.8	1.180	0.8471	169.4	633.8	464.4	0.3413	1.1538	112
113	258.4	243.7	1.163	0.8600	170.6	633.9	463.3	0.3433	1.1524	113
114	262.2	247.5	1.145	0.8730	171.8	633.9	462.1	0.3453	1.1510	114
115	266.2	251.5	1.128	0.8862	173.0	633.9	460.9	0.3474	1.1497	**115**
116	270.1	255.4	1.112	0.8996	174.2	634.0	459.8	0.3495	1.1483	116
117	274.1	259.4	1.095	0.9132	175.4	634.0	458.6	0.3515	1.1469	117
118	278.2	263.5	1.079	0.9269	176.6	634.0	457.4	0.3535	1.1455	118
119	282.3	267.6	1.063	0.9408	177.8	634.0	456.2	0.3556	1.1441	119
120	286.4	271.7	1.047	0.9549	179.0	634.0	455.0	0.3576	1.1427	**120**
121	290.6	275.9	1.032	0.9692	180.2	634.0	453.8	0.3597	1.1414	121
122	294.8	280.1	1.017	0.9837	181.4	634.0	452.6	0.3618	1.1400	122
123	299.1	284.4	1.002	0.9983	182.6	634.0	451.4	0.3638	1.1386	123
124	303.4	288.7	0.987	1.0132	183.9	634.0	450.1	0.3659	1.1372	124
125	307.8	293.1	0.973	1.028	185.1	634.0	448.9	0.3679	1.1358	**125**

TABLE II.—SATURATED AMMONIA: ABSOLUTE-PRESSURE TABLE

Pressure (abs.), lbs./in.²	Temperature, degrees Fahrenheit	Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content		Latent heat, B.t.u./lb.	Entropy			Pressure (abs.), lbs./in.²
				Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Evaporation, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
p	t	V	$1/V$	h	H	L	s	L/T	S	p
5.0	−63.11	49.31	0.02029	−24.5	588.3	612.8	−0.0599	1.5456	1.4857	**5.0**
5.5	−60.27	45.11	0.02217	−21.5	589.5	611.0	−0.0524	1.5301	1.4777	5.5
6.0	−57.64	41.59	0.02405	−18.7	590.6	609.3	−0.0455	1.5158	1.4703	6.0
6.5	−55.18	38.59	0.02591	−16.1	591.6	607.7	−0.0390	1.5026	1.4636	6.5
7.0	−52.88	36.01	0.02777	−13.7	592.5	606.2	−0.0330	1.4904	1.4574	7.0
7.5	−50.70	33.77	0.02962	−11.3	593.4	604.7	−0.0274	1.4790	1.4516	**7.5**
8.0	−48.64	31.79	0.03146	− 9.2	594.2	603.4	−0.0221	1.4683	1.4462	8.0
8.5	−46.69	30.04	0.03329	− 7.1	595.0	602.1	−0.0171	1.4582	1.4411	8.5
9.0	−44.83	28.48	0.03511	− 5.1	595.7	600.8	−0.0123	1.4486	1.4363	9.0
9.5	−43.05	27.08	0.03693	− 3.2	596.4	599.6	−0.0077	1.4396	1.4319	9.5
10.0	−41.34	25.81	0.03874	− 1.4	597.1	598.5	−0.0034	1.4310	1.4276	**10.0**
10.5	−39.71	24.66	0.04055	+ 0.3	597.7	597.4	+0.0007	1.4228	1.4235	10.5
11.0	−38.14	23.61	0.04235	2.0	598.3	596.3	0.0047	1.4149	1.4196	11.0
11.5	−36.62	22.65	0.04414	3.6	598.9	595.3	0.0085	1.4074	1.4159	11.5
12.0	−35.16	21.77	0.04593	5.1	599.4	594.3	0.0122	1.4002	1.4124	12.0
12.5	−33.74	20.96	0.04772	6.7	600.0	593.3	0.0157	1.3933	1.4090	**12.5**
13.0	−32.37	20.20	0.04950	8.1	600.5	592.4	0.0191	1.3866	1.4057	13.0
13.5	−31.05	19.50	0.05128	9.6	601.0	591.4	0.0225	1.3801	1.4026	13.5
14.0	−29.76	18.85	0.05305	10.9	601.4	590.5	0.0257	1.3739	1.3996	14.0
14.5	−28.51	18.24	0.05482	12.2	601.9	589.7	0.0288	1.3679	1.3967	14.5
15.0	−27.29	17.67	0.05658	13.6	602.4	588.8	0.0318	1.3620	1.3938	**15.0**
15.5	−26.11	17.14	0.05834	14.8	602.8	588.0	0.0347	1.3564	1.3911	15.5
16.0	−24.95	16.64	0.06010	16.0	603.2	587.2	0.0375	1.3510	1.3885	16.0
16.5	−23.83	16.17	0.06186	17.2	603.6	586.4	0.0403	1.3456	1.3859	16.5
17.0	−22.73	15.72	0.06361	18.4	604.0	585.6	0.0430	1.3405	1.3835	17.0
17.5	−21.66	15.30	0.06535	19.6	604.4	584.8	0.0456	1.3354	1.3810	**17.5**
18.0	−20.61	14.90	0.06710	20.7	604.8	584.1	0.0482	1.3305	1.3787	18.0
18.5	−19.59	14.53	0.06884	21.8	605.1	583.3	0.0507	1.3258	1.3765	18.5
19.0	−18.58	14.17	0.07058	22.9	605.5	582.6	0.0531	1.3211	1.3742	19.0
19.5	−17.60	13.83	0.07232	23.9	605.8	581.9	0.0555	1.3166	1.3721	19.5
20.0	−16.64	13.50	0.07405	25.0	606.2	581.2	0.0578	1.3122	1.3700	**20.0**
20.5	−15.70	13.20	0.07578	26.0	606.5	580.5	0.0601	1.3078	1.3679	20.5
21.0	−14.78	12.90	0.07751	27.0	606.8	579.8	0.0623	1.3036	1.3659	21.0
21.5	−13.87	12.62	0.07924	27.9	607.1	579.2	0.0645	1.2995	1.3640	21.5
22.0	−12.98	12.35	0.08096	28.9	607.4	578.5	0.0666	1.2955	1.3621	22.0
22.5	−12.11	12.09	0.08268	29.8	607.7	577.9	0.0687	1.2915	1.3602	**22.5**
23.0	−11.25	11.85	0.08440	30.8	608.1	577.3	0.0708	1.2876	1.3584	23.0
23.5	−10.41	11.61	0.08612	31.7	608.3	576.6	0.0728	1.2838	1.3566	23.5
24.0	− 9.58	11.39	0.08783	32.6	608.6	576.0	0.0748	1.2801	1.3549	24.0
24.5	− 8.76	11.17	0.08955	33.5	608.9	575.4	0.0768	1.2764	1.3532	24.5
25.0	− 7.96	10.96	0.09126	34.3	609.1	574.8	0.0787	1.2728	1.3515	**25.0**
25.5	− 7.17	10.76	0.09297	35.2	609.4	574.2	0.0805	1.2693	1.3498	25.5
26.0	− 6.39	10.56	0.09468	36.0	609.7	573.7	0.0824	1.2658	1.3482	26.0
26.5	− 5.63	10.38	0.09638	36.8	609.9	573.1	0.0842	1.2625	1.3467	26.5
27.0	− 4.87	10.20	0.09809	37.7	610.2	572.5	0.0860	1.2591	1.3451	27.0
27.5	− 4.13	10.02	0.09979	38.4	610.4	572.0	0.0878	1.2558	1.3436	**27.5**
28.0	− 3.40	9.853	0.1015	39.3	610.7	571.4	0.0895	1.2526	1.3421	28.0
28.5	− 2.68	9.697	0.1032	40.0	610.9	570.9	0.0912	1.2494	1.3406	28.5
29.0	− 1.97	9.531	0.1049	40.8	611.1	570.3	0.0929	1.2463	1.3392	29.0
29.5	− 1.27	9.383	0.1066	41.6	611.4	569.8	0.0945	1.2433	1.3378	29.5
30	− 0.57	9.236	0.1083	42.3	611.6	569.3	0.0962	1.2402	1.3364	**30**
31	+ 0.79	8.955	0.1117	43.8	612.0	568.2	0.0993	1.2343	1.3336	31
32	2.11	8.693	0.1150	45.2	612.4	567.2	0.1024	1.2286	1.3310	32
33	3.40	8.445	0.1184	46.6	612.8	566.2	0.1055	1.2230	1.3285	33
34	4.66	8.211	0.1218	48.0	613.2	565.2	0.1084	1.2176	1.3260	34

TABLE II.—SATURATED AMMONIA: ABSOLUTE-PRESSURE TABLE (*Continued*)

Pressure (abs.), lbs./in.^{2}	Temperature, degrees Fahrenheit	Volume vapor, ft.^{3}/lb.	Density vapor, lbs./ft.^{3}	Heat content		Latent heat, B.t.u./lb.	Entropy			Pressure (abs.), lbs./in.^{2}
				Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Evaporation, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
p	t	V	$1/V$	h	H	L	s	L/T	S	p
35	5.89	7.991	0.1251	49.3	613.6	564.3	0.1113	1.2123	1.3236	**35**
36	7.09	7.782	0.1285	50.6	614.0	563.4	0.1141	1.2072	1.3213	36
37	8.27	7.584	0.1319	51.9	614.3	562.4	0.1168	1.2022	1.3190	37
38	9.42	7.396	0.1352	53.2	614.7	561.5	0.1195	1.1973	1.3168	38
39	10.55	7.217	0.1386	54.4	615.0	560.6	0.1221	1.1925	1.3146	39
40	11.66	7.047	0.1419	55.6	615.4	559.8	0.1246	1.1879	1.3125	**40**
41	12.74	6.885	0.1452	56.8	615.7	558.9	0.1271	1.1833	1.3104	41
42	13.81	6.731	0.1486	57.9	616.0	558.1	0.1296	1.1788	1.3084	42
43	14.85	6.583	0.1519	59.1	616.3	557.2	0.1320	1.1745	1.3065	43
44	15.88	6.442	0.1552	60.2	616.6	556.4	0.1343	1.1703	1.3046	44
45	16.88	6.307	0.1586	61.3	616.9	555.6	0.1366	1.1661	1.3027	**45**
46	17.87	6.177	0.1619	62.4	617.2	554.8	0.1389	1.1620	1.3009	46
47	18.84	6.053	0.1652	63.4	617.4	554.0	0.1411	1.1580	1.2991	47
48	19.80	5.934	0.1685	64.5	617.7	553.2	0.1433	1.1540	1.2973	48
49	20.74	5.820	0.1718	65.5	618.0	552.5	0.1454	1.1502	1.2956	49
50	21.67	5.710	0.1751	66.5	618.2	551.7	0.1475	1.1464	1.2939	**50**
51	22.58	5.604	0.1785	67.5	618.5	551.0	0.1496	1.1427	1.2923	51
52	23.48	5.502	0.1818	68.5	618.7	550.2	0.1516	1.1390	1.2906	52
53	24.36	5.404	0.1851	69.5	619.0	549.5	0.1536	1.1354	1.2890	53
54	25.23	5.309	0.1884	70.4	619.2	548.8	0.1556	1.1319	1.2875	54
55	26.09	5.218	0.1917	71.4	619.4	548.0	0.1575	1.1284	1.2859	**55**
56	26.94	5.129	0.1950	72.3	619.7	547.4	0.1594	1.1250	1.2844	56
57	27.77	5.044	0.1983	73.3	619.9	546.6	0.1613	1.1217	1.2830	57
58	28.59	4.962	0.2015	74.2	620.1	545.9	0.1631	1.1184	1.2815	58
59	29.41	4.882	0.2048	75.0	620.3	545.3	0.1650	1.1151	1.2801	59
60	30.21	4.805	0.2081	75.9	620.5	544.6	0.1668	1.1119	1.2787	**60**
61	31.00	4.730	0.2114	76.8	620.7	543.9	0.1685	1.1088	1.2773	61
62	31.78	4.658	0.2147	77.7	620.9	543.2	0.1703	1.1056	1.2759	62
63	32.55	4.588	0.2180	78.5	621.1	542.6	0.1720	1.1026	1.2746	63
64	33.31	4.519	0.2213	79.4	621.3	541.9	0.1737	1.0996	1.2733	64
65	34.06	4.453	0.2245	80.2	621.5	541.3	0.1754	1.0966	1.2720	**65**
66	34.81	4.389	0.2278	81.0	621.7	540.7	0.1770	1.0937	1.2707	66
67	35.54	4.327	0.2311	81.8	621.9	540.1	0.1787	1.0907	1.2694	67
68	36.27	4.267	0.2344	82.6	622.0	539.4	0.1803	1.0879	1.2682	68
69	36.99	4.208	0.2377	83.4	622.2	538.8	0.1819	1.0851	1.2670	69
70	37.70	4.151	0.2409	84.2	622.4	538.2	0.1835	1.0823	1.2658	**70**
71	38.40	4.095	0.2442	85.0	622.6	537.6	0.1850	1.0795	1.2645	71
72	39.09	4.041	0.2475	85.8	622.8	537.0	0.1866	1.0768	1.2634	72
73	39.78	3.988	0.2507	86.5	622.9	536.4	0.1881	1.0741	1.2622	73
74	40.46	3.937	0.2540	87.3	623.1	535.8	0.1896	1.0715	1.2611	74
75	41.13	3.887	0.2573	88.0	623.2	535.2	0.1910	1.0689	1.2599	**75**
76	41.80	3.838	0.2606	88.8	623.4	534.6	0.1925	1.0663	1.2588	76
77	42.46	3.790	0.2638	89.5	623.5	534.0	0.1940	1.0637	1.2577	77
78	43.11	3.744	0.2671	90.2	623.7	533.5	0.1954	1.0612	1.2566	78
79	43.76	3.699	0.2704	90.9	623.8	532.9	0.1968	1.0587	1.2555	79
80	44.40	3.655	0.2736	91.7	624.0	532.3	0.1982	1.0563	1.2545	**80**
81	45.03	3.612	0.2769	92.4	624.1	531.7	0.1996	1.0538	1.2534	81
82	45.66	3.570	0.2801	93.1	624.3	531.2	0.2010	1.0514	1.2524	82
83	46.28	3.528	0.2834	93.8	624.4	530.6	0.2024	1.0490	1.2514	83
84	46.89	3.488	0.2867	94.5	624.6	530.1	0.2037	1.0467	1.2504	84
85	47.50	3.449	0.2899	95.1	624.7	529.6	0.2051	1.0443	1.2494	**85**
86	48.11	3.411	0.2932	95.8	624.8	529.0	0.2064	1.0420	1.2484	86
87	48.71	3.373	0.2964	96.5	625.0	528.5	0.2077	1.0397	1.2474	87
88	49.30	3.337	0.2997	97.2	625.1	527.9	0.2090	1.0375	1.2465	88
89	49.89	3.301	0.3030	97.8	625.2	527.4	0.2103	1.0352	1.2455	89

TABLE II.—SATURATED AMMONIA: ABSOLUTE-PRESSURE TABLE (*Continued*)

Pressure (abs.), lbs./in.²	Temperature, degrees Fahrenheit	Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content		Latent heat, B.t.u./lb.	Entropy			Pressure (abs.), lbs./in.²
				Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Evaporation, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
p	t	V	$1/V$	h	H	L	s	L/T	S	p
90	50.47	3.266	0.3062	98.4	625.3	526.9	0.2115	1.0330	1.2445	**90**
91	51.05	3.231	0.3095	99.1	625.5	526.4	0.2128	1.0308	1.2436	91
92	51.62	3.198	0.3127	99.8	625.6	525.8	0.2141	1.0286	1.2427	92
93	52.19	3.165	0.3160	100.4	625.7	525.3	0.2153	1.0265	1.2418	93
94	52.76	3.132	0.3192	101.0	625.8	524.8	0.2165	1.0243	1.2408	94
95	53.32	3.101	0.3225	101.6	625.9	524.3	0.2177	1.0222	1.2399	**95**
96	53.87	3.070	0.3258	102.3	626.1	523.8	0.2190	1.0201	1.2391	96
97	54.42	3.039	0.3290	102.9	626.2	523.3	0.2201	1.0181	1.2382	97
98	54.97	3.010	0.3323	103.5	626.3	522.8	0.2213	1.0160	1.2373	98
99	55.51	2.980	0.3355	104.1	626.4	522.3	0.2225	1.0140	1.2365	99
100	56.05	2.952	0.3388	104.7	626.5	521.8	0.2237	1.0119	1.2356	**100**
102	57.11	2.896	0.3453	105.9	626.7	520.8	0.2260	1.0079	1.2339	102
104	58.16	2.843	0.3518	107.1	626.9	519.8	0.2282	1.0041	1.2323	104
106	59.19	2.791	0.3583	108.3	627.1	518.8	0.2305	1.0002	1.2307	106
108	60.21	2.741	0.3648	109.4	627.3	517.9	0.2327	0.9964	1.2291	108
110	61.21	2.693	0.3713	110.5	627.5	517.0	0.2348	0.9927	1.2275	**110**
112	62.20	2.647	0.3778	111.7	627.7	516.0	0.2369	0.9890	1.2259	112
114	63.17	2.602	0.3843	112.8	627.9	515.1	0.2390	0.9854	1.2244	114
116	64.13	2.559	0.3909	113.9	628.1	514.2	0.2411	0.9819	1.2230	116
118	65.08	2.517	0.3974	114.9	628.2	513.2	0.2431	0.9784	1.2215	118
120	66.02	2.476	0.4039	116.0	628.4	512.4	0.2452	0.9749	1.2201	**120**
122	66.94	2.437	0.4104	117.1	628.6	511.5	0.2471	0.9715	1.2186	122
124	67.86	2.399	0.4169	118.1	628.7	510.6	0.2491	0.9682	1.2173	124
126	68.76	2.362	0.4234	119.1	628.9	509.8	0.2510	0.9649	1.2159	126
128	69.65	2.326	0.4299	120.1	629.0	508.9	0.2529	0.9616	1.2145	128
130	70.53	2.291	0.4364	121.1	629.2	508.1	0.2548	0.9584	1.2132	**130**
132	71.40	2.258	0.4429	122.1	629.3	507.2	0.2567	0.9552	1.2119	132
134	72.26	2.225	0.4494	123.1	629.5	506.4	0.2585	0.9521	1.2106	134
136	73.11	2.193	0.4559	124.1	629.6	505.5	0.2603	0.9490	1.2093	136
138	73.95	2.162	0.4624	125.1	629.8	504.7	0.2621	0.9460	1.2081	138
140	74.79	2.132	0.4690	126.0	629.9	503.9	0.2638	0.9430	1.2068	**140**
142	75.61	2.103	0.4755	126.9	630.0	503.1	0.2656	0.9400	1.2056	142
144	76.42	2.075	0.4820	127.9	630.2	502.3	0.2673	0.9371	1.2044	144
146	77.23	2.047	0.4885	128.8	630.3	501.5	0.2690	0.9342	1.2032	146
148	78.03	2.020	0.4951	129.7	630.4	500.7	0.2707	0.9313	1.2020	148
150	78.81	1.994	0.5016	130.6	630.5	499.9	0.2724	0.9285	1.2009	**150**
152	79.60	1.968	0.5081	131.5	630.6	499.1	0.2740	0.9257	1.1997	152
154	80.37	1.943	0.5147	132.4	630.7	498.3	0.2756	0.9229	1.1985	154
156	81.13	1.919	0.5212	133.3	630.9	497.6	0.2772	0.9202	1.1974	156
158	81.89	1.895	0.5277	134.2	631.0	496.8	0.2788	0.9175	1.1963	158
160	82.64	1.872	0.5343	135.0	631.1	496.1	0.2804	0.9148	1.1952	**160**
162	83.39	1.849	0.5408	135.9	631.2	495.3	0.2820	0.9122	1.1942	162
164	84.12	1.827	0.5473	136.8	631.3	494.5	0.2835	0.9096	1.1931	164
166	84.85	1.805	0.5539	137.6	631.4	493.8	0.2850	0.9070	1.1920	166
168	85.57	1.784	0.5604	138.4	631.5	493.1	0.2866	0.9044	1.1910	168
170	86.29	1.764	0.5670	139.3	631.6	492.3	0.2881	0.9019	1.1900	**170**
172	87.00	1.744	0.5735	140.1	631.7	491.6	0.2895	0.8994	1.1889	172
174	87.71	1.724	0.5801	140.9	631.7	490.8	0.2910	0.8969	1.1879	174
176	88.40	1.705	0.5866	141.7	631.8	490.1	0.2925	0.8944	1.1869	176
178	89.10	1.686	0.5932	142.5	631.9	489.4	0.2939	0.8920	1.1859	178
180	89.78	1.667	0.5998	143.3	632.0	488.7	0.2954	0.8896	1.1850	**180**
182	90.46	1.649	0.6063	144.1	632.1	488.0	0.2968	0.8872	1.1840	182
184	91.14	1.632	0.6129	144.8	632.1	487.3	0.2982	0.8848	1.1830	184
186	91.80	1.614	0.6195	145.6	632.2	486.6	0.2996	0.8825	1.1821	186
188	92.47	1.597	0.6261	146.4	632.3	485.9	0.3010	0.8801	1.1811	188

TABLE II.—SATURATED AMMONIA: ABSOLUTE-PRESSURE TABLE (*Continued*)

Pressure (abs.), lbs./in.²	Temperature, degrees Fahrenheit	Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content		Latent heat, B.t.u./lb.	Entropy			Pressure (abs.), lbs./in.²
				Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Evaporation, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
p	t	V	$1/V$	h	H	L	s	L/T	S	p
190	93.13	1.581	0.6326	147.2	632.4	485.2	0.3024	0.8778	1.1802	**190**
192	93.78	1.564	0.6392	147.9	632.4	484.5	0.3037	0.8755	1.1792	192
194	94.43	1.548	0.6458	148.7	632.5	483.8	0.3050	0.8733	1.1783	194
196	95.07	1.533	0.6524	149.5	632.6	483.1	0.3064	0.8710	1.1774	196
198	95.71	1.517	0.6590	150.2	632.6	482.4	0.3077	0.8688	1.1765	198
200	96.34	1.502	0.6656	150.9	632.7	481.8	0.3090	0.8666	1.1756	**200**
205	97.90	1.466	0.6821	152.7	632.8	480.1	0.3122	0.8612	1.1734	205
210	99.43	1.431	0.6986	154.6	633.0	478.4	0.3154	0.8559	1.1713	210
215	100.94	1.398	0.7152	156.3	633.1	476.8	0.3185	0.8507	1.1692	215
220	102.42	1.367	0.7318	158.0	633.2	475.2	0.3216	0.8455	1.1671	220
225	103.87	1.336	0.7484	159.7	633.3	473.6	0.3246	0.8405	1.1651	**225**
230	105.30	1.307	0.7650	161.4	633.4	472.0	0.3275	0.8356	1.1631	230
235	106.71	1.279	0.7817	163.1	633.5	470.4	0.3304	0.8307	1.1611	235
240	108.09	1.253	0.7984	164.7	633.6	468.9	0.3332	0.8260	1.1592	240
245	109.46	1.227	0.8151	166.4	633.7	467.3	0.3360	0.8213	1.1573	245
250	110.80	1.202	0.8319	168.0	633.8	465.8	0.3388	0.8167	1.1555	**250**
255	112.12	1.178	0.8487	169.5	633.8	464.3	0.3415	0.8121	1.1536	255
260	113.42	1.155	0.8655	171.1	633.9	462.8	0.3441	0.8077	1.1518	260
265	114.71	1.133	0.8824	172.6	633.9	461.3	0.3468	0.8033	1.1501	265
270	115.97	1.112	0.8993	174.1	633.9	459.8	0.3494	0.7989	1.1483	270
275	117.22	1.091	0.9162	175.6	634.0	458.4	0.3519	0.7947	1.1466	**275**
280	118.45	1.072	0.9332	177.1	634.0	456.9	0.3545	0.7904	1.1449	280
285	119.66	1.052	0.9502	178.6	634.0	455.4	0.3569	0.7863	1.1432	285
290	120.86	1.034	0.9672	180.0	634.0	454.0	0.3594	0.7821	1.1415	290
295	122.05	1.016	0.9843	181.5	634.0	452.5	0.3618	0.7781	1.1399	295
300	123.21	0.999	1.0015	182.9	634.0	451.1	0.3642	0.7741	1.1383	**300**

TABLE III.—SATURATED AMMONIA: GAGE-PRESSURE TABLE

Pressure (gage), lbs./in.²	Temperature degrees Fahrenheit	Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content		Latent heat, B.t.u./lb.	Entropy			Pressure (gage), lbs./in.²
				Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Evaporation, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
g. p.	t	V	1/V	h	H	L	s	L/T	S	g. p.
20*	−63.9	50.5	0.0198	−25.3	588.0	613.3	−0.062	1.550	1.488	**20***
19*	−61.0	46.2	0.0217	−22.3	589.2	611.5	−0.055	1.535	1.480	19*
18*	−58.4	42.6	0.0235	−19.5	590.3	609.8	−0.048	1.521	1.473	18*
17*	−55.9	39.5	0.0253	−16.9	591.3	608.2	−0.041	1.507	1.466	17*
16*	−53.6	36.8	0.0272	−14.5	592.2	606.7	−0.035	1.495	1.460	16*
15*	−51.4	34.5	0.0290	−12.2	593.1	605.3	−0.029	1.483	1.454	**15***
14*	−49.4	32.5	0.0308	−10.0	593.9	603.9	−0.023	1.472	1.449	14*
13*	−47.4	30.7	0.0326	− 7.9	594.7	602.6	−0.019	1.462	1.443	13*
12*	−45.6	29.1	0.0344	− 5.9	595.4	601.3	−0.014	1.452	1.438	12*
11*	−43.8	27.6	0.0362	− 4.0	596.1	600.1	−0.010	1.443	1.433	11*
10*	−42.1	26.3	0.0380	− 2.2	596.8	599.0	−0.005	1.434	1.429	**10***
9*	−40.4	25.2	0.0397	− 0.5	597.4	597.9	−0.001	1.426	1.425	9*
8*	−38.9	24.1	0.0415	+ 1.2	598.0	596.8	+0.003	1.418	1.421	8*
7*	−37.3	23.1	0.0433	2.8	598.6	595.8	0.007	1.411	1.418	7*
6*	−35.9	22.2	0.0450	4.4	599.1	594.7	0.010	1.405	1.415	6*
5*	−34.5	21.4	0.0468	5.9	599.6	593.7	0.014	1.397	1.411	**5***
4*	−33.1	20.6	0.0485	7.4	600.2	592.8	0.017	1.390	1.407	4*
3*	−31.8	19.9	0.0503	8.8	600.7	591.9	0.020	1.384	1.404	3*
2*	−30.5	19.2	0.0520	10.2	601.2	591.0	0.024	1.377	1.401	2*
1*	−29.2	18.6	0.0538	11.5	601.6	590.1	0.027	1.371	1.398	1*
0	−28.0	18.0	0.0555	12.8	602.1	589.3	0.030	1.366	1.396	**0**
1	−25.6	16.9	0.0590	15.4	603.0	587.6	0.036	1.354	1.390	1
2	−23.4	16.0	0.0626	17.8	603.8	586.0	0.041	1.344	1.385	2
3	−21.2	15.1	0.0661	20.1	604.6	584.5	0.047	1.333	1.380	3
4	−19.2	14.4	0.0695	22.3	605.3	583.0	0.052	1.324	1.376	4
5	−17.2	13.7	0.0730	24.4	606.0	581.6	0.056	1.315	1.371	**5**
6	−15.3	13.1	0.0765	26.4	606.6	580.2	0.061	1.306	1.367	6
7	−13.5	12.5	0.0799	28.4	607.3	578.9	0.065	1.298	1.363	7
8	−11.8	12.0	0.0834	30.3	607.9	577.6	0.070	1.290	1.360	8
9	−10.1	11.5	0.0868	32.1	608.4	576.3	0.074	1.282	1.356	9
10	− 8.4	11.1	0.0902	33.8	609.0	575.2	0.078	1.275	1.353	**10**
11	− 6.9	10.7	0.0937	35.5	609.5	574.0	0.081	1.268	1.349	11
12	− 5.3	10.3	0.0971	37.1	610.0	572.9	0.085	1.261	1.346	12
13	− 3.8	9.96	0.100	38.8	610.5	571.7	0.088	1.255	1.343	13
14	− 2.4	9.63	0.104	40.4	611.0	570.6	0.092	1.248	1.340	14
15	− 1.0	9.32	0.107	41.9	611.4	569.5	0.095	1.242	1.337	**15**
16	+ 0.4	9.04	0.111	43.4	611.9	568.5	0.098	1.236	1.334	16
17	1.7	8.78	0.114	44.8	612.3	567.5	0.101	1.230	1.331	17
18	3.0	8.53	0.117	46.2	612.7	566.5	0.104	1.225	1.329	18
19	4.3	8.28	0.121	47.6	613.1	565.5	0.107	1.219	1.326	19
20	5.5	8.06	0.124	48.9	613.5	564.6	0.110	1.214	1.324	**20**
21	6.7	7.85	0.127	50.2	613.9	563.7	0.113	1.209	1.322	21
22	7.9	7.65	0.131	51.5	614.2	562.7	0.116	1.204	1.320	22
23	9.1	7.46	0.134	52.8	614.6	561.8	0.119	1.199	1.318	23
24	10.2	7.28	0.138	54.0	614.9	560.9	0.121	1.194	1.315	24
25	11.3	7.11	0.141	55.3	615.3	560.0	0.124	1.189	1.313	**25**
26	12.4	6.94	0.144	56.5	615.6	559.1	0.126	1.185	1.311	26
27	13.5	6.78	0.148	57.6	615.9	558.3	0.129	1.180	1.309	27
28	14.5	6.63	0.151	58.8	616.2	557.4	0.131	1.176	1.307	28
29	15.6	6.49	0.154	59.9	616.5	556.6	0.134	1.171	1.305	29
30	16.6	6.35	0.158	61.0	616.8	555.8	0.136	1.167	1.303	**30**
31	17.6	6.22	0.161	62.1	617.1	555.0	0.138	1.163	1.301	31
32	18.6	6.09	0.164	63.2	617.4	554.2	0.140	1.159	1.299	32
33	19.5	5.97	0.168	64.2	617.6	553.4	0.143	1.155	1.298	33
34	20.5	5.85	0.171	65.3	617.9	552.6	0.145	1.151	1.296	34

* Inches of mercury below 1 standard atmosphere (29.92 inches).

TABLE III.—SATURATED AMMONIA: GAGE-PRESSURE TABLE (*Continued*)

Pressure (gage), lbs./in.²	Temperature, degrees Fahrenheit	Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content: Liquid, B.t.u./lb.	Heat content: Vapor, B.t.u./lb.	Latent heat, B.t.u./lb.	Entropy: Liquid, B.t.u./lb. degrees Fahrenheit	Entropy: Evaporation, B.t.u./lb. degrees Fahrenheit	Entropy: Vapor, B.t.u./lb. degrees Fahrenheit	Pressure (gage), lbs./in.²
g. p.	*t*	*V*	*1/V*	*h*	*H*	*L*	*s*	*L/T*	*S*	*g. p.*
35	21.4	5.74	0.174	66.3	618.2	551.9	0.147	1.148	1.295	**35**
36	22.3	5.64	0.177	67.3	618.4	551.1	0.149	1.144	1.293	36
37	23.2	5.54	0.181	68.3	618.7	550.4	0.151	1.140	1.291	37
38	24.1	5.44	0.184	69.2	618.9	549.7	0.153	1.137	1.290	38
39	25.0	5.34	0.187	70.2	619.1	548.9	0.155	1.133	1.288	39
40	25.8	5.25	0.191	71.2	619.4	548.2	0.157	1.130	1.287	**40**
41	26.7	5.16	0.194	72.1	619.6	547.5	0.159	1.126	1.285	41
42	27.5	5.07	0.197	73.0	619.8	546.8	0.161	1.123	1.284	42
43	28.3	4.99	0.201	73.9	620.0	546.1	0.163	1.119	1.282	43
44	29.2	4.91	0.204	74.8	620.3	545.5	0.164	1.116	1.280	44
45	30.0	4.83	0.207	75.7	620.5	544.8	0.166	1.113	1.279	**45**
46	30.8	4.76	0.210	76.6	620.7	544.1	0.168	1.110	1.278	46
47	31.5	4.68	0.214	77.4	620.9	543.5	0.170	1.107	1.277	47
48	32.3	4.61	0.217	78.3	621.1	542.8	0.171	1.104	1.275	48
49	33.1	4.54	0.220	79.1	621.3	542.2	0.173	1.101	1.274	49
50	33.8	4.48	0.224	80.0	621.5	541.5	0.175	1.098	1.273	**50**
51	34.6	4.41	0.227	80.8	621.7	540.9	0.177	1.095	1.272	51
52	35.3	4.35	0.230	81.6	621.8	540.2	0.178	1.092	1.270	52
53	36.1	4.29	0.233	82.4	622.0	539.6	0.180	1.089	1.269	53
54	36.8	4.23	0.237	83.2	622.2	539.0	0.181	1.086	1.267	54
55	37.5	4.17	0.240	84.0	622.4	538.4	0.183	1.083	1.266	**55**
56	38.2	4.12	0.243	84.8	622.5	537.7	0.185	1.080	1.265	56
57	38.9	4.06	0.246	85.6	622.7	537.1	0.186	1.078	1.264	57
58	39.6	4.01	0.250	86.3	622.9	536.6	0.188	1.075	1.263	58
59	40.3	3.96	0.253	87.0	623.0	536.0	0.189	1.072	1.261	59
60	40.9	3.91	0.256	87.8	623.2	535.4	0.191	1.069	1.260	**60**
61	41.6	3.86	0.260	88.6	623.4	534.8	0.192	1.067	1.259	61
62	42.3	3.81	0.263	89.3	623.5	534.2	0.194	1.064	1.258	62
63	42.9	3.77	0.266	90.0	623.7	533.7	0.195	1.062	1.257	63
64	43.6	3.72	0.269	90.7	623.8	533.1	0.196	1.060	1.256	64
65	44.2	3.67	0.273	91.5	624.0	532.5	0.198	1.057	1.255	**65**
66	44.8	3.63	0.276	92.2	624.1	531.9	0.199	1.055	1.254	66
67	45.5	3.59	0.279	92.9	624.2	531.3	0.201	1.052	1.253	67
68	46.1	3.55	0.282	93.6	624.4	530.8	0.202	1.050	1.252	68
69	46.7	3.51	0.286	94.3	624.5	530.2	0.203	1.048	1.251	69
70	47.3	3.47	0.289	94.9	624.6	529.7	0.205	1.045	1.250	**70**
71	47.9	3.43	0.292	95.6	624.8	529.2	0.206	1.043	1.249	71
72	48.5	3.39	0.295	96.3	624.9	528.6	0.207	1.041	1.248	72
73	49.1	3.35	0.299	97.0	625.1	528.1	0.209	1.038	1.247	73
74	49.7	3.32	0.302	97.6	625.2	527.6	0.210	1.036	1.246	74
75	50.3	3.28	0.305	98.3	625.3	527.0	0.211	1.034	1.245	**75**
76	50.9	3.24	0.308	98.9	625.4	526.5	0.212	1.032	1.244	76
77	51.5	3.21	0.312	99.5	625.5	526.0	0.214	1.029	1.243	77
78	52.0	3.17	0.315	100.2	625.7	525.5	0.215	1.027	1.242	78
79	52.6	3.14	0.318	100.8	625.8	525.0	0.216	1.025	1.241	79
80	53.1	3.11	0.322	101.5	625.9	524.4	0.217	1.023	1.240	**80**
81	53.7	3.08	0.325	102.1	626.0	523.9	0.219	1.020	1.239	81
82	54.3	3.05	0.328	102.7	626.1	523.4	0.220	1.018	1.238	82
83	54.8	3.02	0.331	103.3	626.3	523.0	0.221	1.016	1.237	83
84	55.3	2.99	0.335	103.9	626.4	522.5	0.222	1.015	1.237	84
85	55.9	2.96	0.338	104.5	626.5	522.0	0.223	1.013	1.236	**85**
86	56.4	2.94	0.341	105.1	626.6	521.5	0.224	1.011	1.235	86
87	57.0	2.91	0.344	105.7	626.7	521.0	0.226	1.008	1.234	87
88	57.5	2.88	0.348	106.3	626.8	520.5	0.227	1.006	1.233	88
89	58.0	2.85	0.351	106.9	626.9	520.0	0.228	1.005	1.233	89

TABLE III.—SATURATED AMMONIA: GAGE-PRESSURE TABLE (*Continued*)

Pressure (gage), lbs./in.²	Temperature, degrees Fahrenheit	Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content: Liquid, B.t.u./lb.	Heat content: Vapor, B.t.u./lb.	Latent heat, B.t.u./lb.	Entropy: Liquid, B.t.u./lb. degrees Fahrenheit	Entropy: Evaporation, B.t.u./lb. degrees Fahrenheit	Entropy: Vapor, B.t.u./lb. degrees Fahrenheit	Pressure (gage), lbs./in.²
g. p.	*t*	*V*	1/*V*	*h*	*H*	*L*	*s*	*L/T*	*S*	*g. p.*
90	58.5	2.82	0.354	107.5	627.0	519.5	0.229	1.003	1.232	**90**
91	59.0	2.80	0.357	108.1	627.1	519.0	0.230	1.001	1.231	91
92	59.6	2.77	0.361	108.7	627.2	518.5	0.231	0.999	1.230	92
93	60.1	2.75	0.364	109.3	627.3	518.0	0.232	0.997	1.229	93
94	60.6	2.72	0.367	109.8	627.4	517.6	0.233	0.995	1.228	94
95	61.1	2.70	0.370	110.4	627.5	517.1	0.235	0.993	1.228	**95**
96	61.6	2.68	0.374	111.0	627.6	516.6	0.236	0.991	1.227	96
97	62.0	2.65	0.377	111.6	627.7	516.1	0.237	0.989	1.226	97
98	62.5	2.63	0.380	112.1	627.8	515.7	0.238	0.988	1.226	98
99	63.0	2.61	0.383	112.6	627.9	515.3	0.239	0.986	1.225	99
100	63.5	2.59	0.287	113.2	628.0	514.8	0.240	0.984	1.224	**100**
102	64.5	2.54	0.393	114.2	628.1	513.9	0.242	0.981	1.223	102
104	65.4	2.50	0.400	115.3	628.3	513.0	0.244	0.977	1.221	104
106	66.4	2.46	0.406	116.4	628.5	512.1	0.246	0.974	1.220	106
108	67.3	2.42	0.413	117.4	628.6	511.2	0.248	0.970	1.218	108
110	68.2	2.39	0.419	118.5	628.8	510.3	0.250	0.967	1.217	**110**
112	69.1	2.35	0.426	119.5	628.9	509.4	0.252	0.964	1.216	112
114	70.0	2.31	0.432	120.5	629.1	508.6	0.254	0.960	1.214	114
116	70.8	2.28	0.439	121.5	629.3	507.8	0.256	0.957	1.213	116
118	71.7	2.25	0.445	122.5	629.4	506.9	0.257	0.954	1.211	118
120	72.6	2.21	0.452	123.5	629.5	506.0	0.259	0.951	1.210	**120**
122	73.4	2.18	0.458	124.5	629.7	505.2	0.261	0.948	1.209	122
124	74.2	2.15	0.465	125.4	629.8	504.4	0.263	0.945	1.208	124
126	75.1	2.12	0.471	126.3	629.9	503.6	0.364	0.942	1.206	126
128	75.9	2.09	0.478	127.3	630.1	502.8	0.266	0.939	1.205	128
130	76.7	2.06	0.484	128.2	630.2	502.0	0.268	0.936	1.204	**130**
132	77.5	2.04	0.491	129.1	630.3	501.2	0.270	0.933	1.203	132
134	78.3	2.01	0.497	130.0	630.4	500.4	0.271	0.930	1.201	134
136	79.1	1.98	0.504	130.9	630.5	499.6	0.273	0.927	1.200	136
138	79.9	1.96	0.510	131.8	630.7	498.9	0.274	0.925	1.199	138
140	80.6	1.93	0.517	132.7	630.8	498.1	0.276	0.922	1.198	**140**
142	81.4	1.91	0.523	133.6	630.9	497.3	0.278	0.919	1.197	142
144	82.2	1.89	0.530	134.5	631.0	496.5	0.279	0.917	1.196	144
146	82.9	1.86	0.536	135.3	631.1	495.8	0.281	0.914	1.195	146
148	83.6	1.84	0.543	136.2	631.2	495.0	0.283	0.911	1.194	148
150	84.4	1.82	0.550	137.0	631.3	494.3	0.284	0.909	1.193	**150**
152	85.1	1.80	0.556	137.9	631.4	493.5	0.286	0.906	1.192	152
154	85.8	1.78	0.563	138.7	631.5	492.8	0.287	0.904	1.191	154
156	86.5	1.76	0.569	139.5	631.6	492.1	0.289	0.901	1.190	156
158	87.2	1.74	0.576	140.3	631.7	491.4	0.290	0.899	1.189	158
160	88.0	1.72	0.582	141.1	631.8	490.7	0.292	0.896	1.188	**160**
162	88.6	1.70	0.589	141.9	631.9	490.0	0.293	0.894	1.187	162
164	89.3	1.68	0.595	142.7	631.9	489.2	0.294	0.891	1.185	164
166	90.0	1.66	0.602	143.5	632.0	488.5	0.296	0.889	1.185	166
168	90.7	1.64	0.609	144.3	632.1	487.8	0.297	0.886	1.183	168
170	91.4	1.62	0.615	145.1	632.1	487.0	0.299	0.884	1.183	**170**
172	92.0	1.61	0.622	145.8	632.2	486.4	0.300	0.882	1.182	172
174	92.7	1.59	0.628	146.6	632.3	485.7	0.302	0.879	1.181	174
176	93.4	1.57	0.635	147.4	632.4	485.0	0.303	0.877	1.180	176
178	94.0	1.56	0.641	148.2	632.5	484.3	0.304	0.875	1.179	178
180	94.7	1.54	0.648	148.9	632.5	483.6	0.305	0.873	1.178	**180**
182	95.3	1.53	0.655	149.7	632.6	482.9	0.307	0.870	1.177	182
184	95.9	1.51	0.661	150.5	632.7	482.2	0.308	0.868	1.176	184
186	96.6	1.50	0.668	151.2	632.7	481.5	0.309	0.866	1.175	186
188	97.2	1.48	0.674	151.9	632.8	480.9	0.311	0.863	1.174	188

TABLE III.—SATURATED AMMONIA: GAGE-PRESSURE TABLE (*Continued*)

Pressure (gage), lbs./in.²	Temperature, degrees Fahrenheit	Volume vapor, ft.³/lb.	Density vapor, lbs./ft.³	Heat content		Latent heat, B.t.u./lb.	Entropy			Pressure (gage), lbs./in.²
				Liquid, B.t.u./lb.	Vapor, B.t.u./lb.		Liquid, B.t.u./lb. degrees Fahrenheit	Evaporation, B.t.u./lb. degrees Fahrenheit	Vapor, B.t.u./lb. degrees Fahrenheit	
g. p.	*t*	*V*	*1/V*	*h*	*H*	*L*	*s*	*L/T*	*S*	*g. p.*
190	97.8	1.47	0.681	152.6	632.8	480.2	0.312	0.861	1.173	**190**
192	98.4	1.45	0.688	153.4	632.9	479.5	0.314	0.859	1.173	192
194	99.0	1.44	0.694	154.0	632.9	478.9	0.315	0.857	1.172	194
196	99.7	1.43	0.701	154.8	633.0	478.2	0.316	0.855	1.171	196
198	100.3	1.41	0.708	155.5	633.0	477.5	0.317	0.853	1.170	198
200	100.9	1.40	0.714	156.2	633.1	476.9	0.318	0.851	1.169	**200**
205	102.3	1.37	0.731	158.0	633.2	475.2	0.321	0.846	1.167	205
210	103.8	1.34	0.747	159.6	633.3	473.7	0.324	0.841	1.165	210
215	105.2	1.31	0.764	161.3	633.4	472.1	0.327	0.836	1.163	215
220	106.6	1.28	0.781	163.0	633.5	470.5	0.330	0.831	1.161	220
225	108.0	1.25	0.797	164.6	633.6	469.0	0.333	0.826	1.159	**225**
230	109.4	1.23	0.814	166.3	633.7	467.4	0.336	0.822	1.158	230
235	110.7	1.20	0.831	167.9	633.8	465.9	0.339	0.817	1.156	235
240	112.0	1.18	0.848	169.4	633.8	464.4	0.341	0.813	1.154	240
245	113.3	1.16	0.864	171.0	633.9	462.9	0.344	0.808	1.152	245
250	114.6	1.13	0.881	172.6	633.9	461.3	0.346	0.804	1.150	**250**
255	115.9	1.11	0.898	174.1	634.0	459.9	0.349	0.799	1.148	255
260	117.1	1.09	0.915	175.6	634.0	458.4	0.352	0.795	1.147	260
265	118.4	1.07	0.932	177.0	634.0	457.0	0.354	0.791	1.145	265
270	119.6	1.05	0.949	178.5	634.0	455.5	0.357	0.786	1.143	270
275	120.8	1.03	0.966	179.9	634.0	454.1	0.359	0.783	1.142	**275**
280	122.0	1.02	0.983	181.4	634.0	452.6	0.362	0.778	1.140	280
285	123.1	1.00	1.000	182.8	634.0	451.2	0.364	0.774	1.138	285
290	124.3	0.98	1.018	184.2	634.0	449.8	0.367	0.770	1.137	290
295	125.4	0.97	1.035	185.6	634.0	448.4	0.369	0.766	1.135	295
300	126.5	0.95	1.052	187.0	633.9	446.9	0.371	0.762	1.133	**300**

TABLE IV.—PROPERTIES OF LIQUID AMMONIA

Temperature, degrees Fahrenheit	Saturation						Latent heat of pressure variation,	Variation of h with p (t constant),	Compressibility,	Temperature, degrees Fahrenheit
	Pressure (abs.), lbs./in.²	Volume, ft.³/lb.	Density, lbs./ft.³	Specific heat, B.t.u./lb. degrees Fahrenheit	Heat content, B.t.u./lb.	Latent heat, B.t.u./lb.	$\frac{\text{B.t.u./lb.}}{\text{lb./in.}^2}$	$\frac{\text{B.t.u./lb.}}{\text{lb./in.}^2}$	per lb./in.² $\times 10^6$	
t	p	v	$\frac{1}{v}$	c	h	L	l	$\left(\frac{\partial h}{\partial p}\right)_t$	$-\frac{1}{v}\left(\frac{\partial v}{\partial p}\right)_t$	t
Triple point*...	......	0.01961*	51.00							
	0.88	0.02182	45.83	....	....	....	....	....	....	−107.86
−100	1.24	0.02197	45.52	(1.040)	(−63.0)	(633)	....	....	....	**−100**
− 95	1.52	0.02207	45.32	(1.042)	(−57.8)	(631)	....	....	....	− 95
− 90	1.86	0.02216	45.12	(1.043)	(−52.6)	(628)	....	....	....	− 90
− 85	2.27	0.02226	44.92	(1.045)	(−47.4)	(625)	....	....	....	− 85
− 80	2.74	0.02236	44.72	(1.046)	(−42.2)	(622)	....	....	....	− 80
− 75	3.29	0.02246	44.52	(1.048)	(−36.9)	(619)	....	....	....	**− 75**
− 70	3.94	0.02256	44.32	(1.050)	(−31.7)	(616)	....	....	....	− 70
− 65	4.69	0.02267	44.11	(1.052)	(−26.4)	(613)	....	....	....	− 65
− 60	5.55	0.02278	43.91	1.054	−21.18	610.8	−0.0016	0.0026	4.4	− 60
− 55	6.54	0.02288	43.70	1.056	−15.90	607.5	−0.0016	0.0026	4.5	− 55
− 50	7.67	0.02299	43.49	1.058	−10.61	604.3	−0.0017	0.0026	4.6	**− 50**
− 45	8.95	0.02310	43.28	1.060	− 5.31	600.9	−0.0017	0.0026	4.7	− 45
− 40	10.41	0.02322	43.08	1.062	0.00	597.6	−0.0018	0.0025	4.8	− 40
− 35	12.05	0.02333	42.86	1.064	+ 5.32	594.2	−0.0018	0.0025	5.0	− 35
− 30	13.90	0.02345	42.65	1.066	10.66	590.7	−0.0019	0.0025	5.1	− 30
− 25	15.98	0.02357	42.44	1.068	16.00	587.2	−0.0019	0.0024	5.2	**− 25**
− 20	18.30	0.02369	42.22	1.070	21.36	583.6	−0.0020	0.0024	5.4	− 20
− 15	20.88	0.02381	42.00	1.073	26.73	580.0	−0.0020	0.0024	5.5	− 15
− 10	23.74	0.02393	41.78	1.075	32.11	576.4	−0.0021	0.0023	5.7	− 10
− 5	26.92	0.02406	41.56	1.078	37.51	572.6	−0.0022	0.0023	5.8	− 5
0	30.42	0.02419	41.34	1.080	42.92	568.9	−0.0022	0.0022	6.0	**0**
5	34.27	0.02432	41.11	1.083	48.35	565.0	−0.0023	0.0022	6.2	5
10	38.51	0.02446	40.89	1.085	53.79	561.1	−0.0024	0.0021	6.4	10
15	43.14	0.02460	40.66	1.088	59.24	557.1	−0.0025	0.0021	6.6	15
20	48.21	0.02474	40.43	1.091	64.71	553.1	−0.0025	0.0020	6.8	20
25	53.73	0.02488	40.20	1.094	70.20	548.9	−0.0026	0.0020	7.0	**25**
30	59.74	0.02503	39.96	1.097	75.71	544.8	−0.0027	0.0019	7.3	30
35	66.26	0.02518	39.72	1.100	81.23	540.5	−0.0028	0.0019	7.5	35
40	73.32	0.02533	39.49	1.104	86.77	536.2	−0.0029	0.0018	7.8	40
45	80.96	0.02548	39.24	1.108	92.34	531.8	−0.0030	0.0017	8.1	45

50	89.19	0.02564	39.00	1.112	97.93	527.3	−0.0031	0.0017	8.4	**50**
55	98.06	0.02581	38.75	1.116	103.54	522.8	−0.0032	0.0016	8.8	55
60	107.6	0.02597	38.50	1.120	109.18	518.1	−0.0033	0.0015	9.1	60
65	117.8	0.02614	38.25	1.125	114.85	513.4	−0.0034	0.0014	9.5	65
70	128.8	0.02632	38.00	1.129	120.54	508.6	−0.0035	0.0013	10.0	70
75	140.5	0.02650	37.74	1.133	126.25	503.7	−0.0037	0.0012	10.4	**75**
80	153.0	0.02668	37.48	1.138	131.99	498.7	−0.0038	0.0011	10.9	80
85	166.4	0.02687	37.21	1.142	137.75	493.6	−0.0040	0.0010	11.4	85
90	180.6	0.02707	36.95	1.147	143.54	488.5	−0.0041	0.0009	12.0	90
95	195.8	0.02727	36.67	1.151	149.36	483.2	−0.0043	0.0008	12.6	95
100	211.9	0.02747	36.40	1.156	155.21	477.8	−0.0045	0.0006	13.3	**100**
105	228.9	0.02769	36.12	1.162	161.09	472.3	−0.0047	0.0005	14.1	105
110	247.0	0.02790	35.84	1.168	167.01	466.7	−0.0049	0.0003	14.9	110
115	266.2	0.02813	35.55	1.176	172.97	460.9	−0.0051	0.0001	15.8	115
120	286.4	0.02836	35.26	1.183	178.98	455.0	−0.0053	0.0000	16.7	120
125	307.8	0.02860	34.96	(1.189)	(185)	(449)	....	......	....	**125**
130	330.3	0.02885	34.66	(1.197)	(191)	(443)	....	......	....	130
135	354.1	0.02911	34.35	(1.205)	(197)	(436)	....	......	....	135
140	379.1	0.02938	34.04	(1.213)	(203)	(430)	....	......	....	140
145	405.5	0.02966	33.72	(1.222)	(210)	(423)	....	......	....	145
150	433.2	0.02995	33.39	(1.23)	(216)	(416)	....	......	....	**150**
155	462.3	0.03025	33.06	(1.24)	(222)	(409)	....	......	....	155
160	492.8	0.03056	32.72	(1.25)	(229)	(401)	....	......	....	160
165	524.8	0.03089	32.37	(1.26)	(235)	(394)	....	......	....	165
170	558.4	0.03124	32.01	(1.27)	(241)	(386)	....	......	....	170
175	593.5	0.03160	31.65	(1.29)	(248)	(377)	....	......	....	**175**
180	630.3	0.03198	31.27	(1.30)	(255)	(369)	....	......	....	180
185	668.7	0.03238	30.88	(1.32)	(262)	(360)	....	......	....	185
190	708.9	0.03281	30.48	(1.34)	(269)	(351)	....	......	....	190
195	750.9	0.03326	30.06	(1.36)	(276)	(342)	....	......	....	195
200	794.7	0.03375	29.63	(1.38)	(283)	(332)	....	......	....	**200**
210	888.1	0.03482	28.72	(1.43)	(297)	(310)	....	......	....	210
220	989.5	0.0361	27.7	(1.49)	(313)	(287)	....	......	....	220
230	1,099.5	0.0376	26.6	(1.57)	(329)	(260)	....	......	....	230
240	1,218.5	0.0395	25.3	(1.70)	(346)	(229)	....	......	....	240
250	1,347	0.0422	23.7	(1.90)	(365)	(192)	....	......	....	250
260	1,486	0.0463	21.6	(2.33)	(387)	(142)	....	......	....	260
270	1,635	0.0577	17.3	(5.30)	(419)	(52)	....	......	....	270
Critical........	1,657	0.0686	14.6	∞	(433)	0	− ∞	− ∞	∞	271.4

* Properties of solid ammonia at the triple point (−107.86° F.).

NOTE.—The figures in parentheses were calculated from empirical equations given in *Bur. Standards Sci. Papers* 313 and 315 and represent values obtained by extrapolation beyond the range covered in the experimental work.

Table V.—Properties of Superheated Ammonia Vapor

V = volume in cubic feet per pound; H = heat content in B.t.u. per pound; S = entropy in B.t.u. per pound, degrees Fahrenheit

Temperature, degrees Fahrenheit	Absolute pressure in pounds per square inch (saturation temperature in italics)											
	15 *−27.29°*			16 *−24.95°*			17 *−22.73°*			18 *−20.61°*		
	V	H	S	V	H	S	V	H	S	V	H	S
Saturation	*17.67*	*602.4*	*1.3938*	*16.64*	*603.2*	*1.3885*	*15.72*	*604.0*	*1.3835*	*14.90*	*604.8*	*1.3787*
−20	18.01	606.4	1.4031	16.86	606.0	1.3948	15.83	605.6	1.3870	14.93	605.1	1.3795
−10	18.47	611.9	1.4154	17.29	611.5	1.4072	16.24	611.1	1.3994	15.32	610.7	1.3921
0	18.92	617.2	1.4272	17.72	616.9	1.4191	16.65	616.6	1.4114	15.70	616.2	1.4042
10	19.37	622.5	1.4386	18.14	622.2	1.4306	17.05	621.9	1.4230	16.08	621.6	1.4158
20	19.82	627.8	1.4497	18.56	627.5	1.4417	17.45	627.2	1.4342	16.46	626.9	1.4270
30	20.26	633.0	1.4604	18.97	632.7	1.4525	17.84	632.5	1.4450	16.83	632.2	1.4380
40	20.70	638.2	1.4709	19.39	638.0	1.4630	18.23	637.7	1.4556	17.20	637.5	1.4486
50	21.14	643.4	1.4812	19.80	643.2	1.4733	18.62	642.9	1.4659	17.57	642.7	1.4590
60	21.58	648.5	1.4912	20.21	648.3	1.4834	19.01	648.1	1.4761	17.94	647.9	1.4691
70	22.01	653.7	1.5011	20.62	653.5	1.4933	19.39	653.3	1.4860	18.30	653.1	1.4790
80	22.44	658.9	1.5108	21.03	658.7	1.5030	19.78	658.5	1.4957	18.67	658.4	1.4887
90	22.88	664.0	1.5203	21.43	663.9	1.5125	20.16	663.7	1.5052	19.03	663.6	1.4983
100	23.31	669.2	1.5296	21.84	669.1	1.5218	20.54	668.9	1.5146	19.39	668.8	1.5077
110	23.74	674.4	1.5388	22.24	674.3	1.5310	20.92	674.1	1.5238	19.75	674.0	1.5169
120	24.17	679.6	1.5478	22.65	679.5	1.5401	21.30	679.3	1.5328	20.11	679.2	1.5260
130	24.60	684.8	1.5567	23.05	684.7	1.5490	21.68	684.5	1.5418	20.47	684.4	1.5349
140	25.03	690.0	1.5655	23.45	689.9	1.5578	22.06	689.8	1.5506	20.83	689.7	1.5438
150	25.46	695.3	1.5742	23.86	695.1	1.5665	22.44	695.0	1.5593	21.19	694.9	1.5525
160	25.88	700.5	1.5827	24.26	700.4	1.5750	22.82	700.3	1.5678	21.54	700.2	1.5610
170	26.31	705.8	1.5911	24.66	705.7	1.5835	23.20	705.6	1.5763	21.90	705.5	1.5695
180	26.74	711.1	1.5995	25.06	711.0	1.5918	23.58	710.9	1.5846	22.26	710.8	1.5778
190	27.16	716.4	1.6077	25.46	716.3	1.6001	23.95	716.2	1.5929	22.61	716.1	1.5861
200	27.59	721.7	1.6158	25.86	721.6	1.6082	24.33	721.5	1.6010	22.97	721.4	1.5943
220	28.44	732.4	1.6318	26.66	732.3	1.6242	25.08	732.2	1.6170	23.68	732.2	1.6103

Temperature, degrees Fahrenheit	19 *−18.58°*			20 *−16.64°*			21 *−14.78°*			22 *−12.98°*		
Saturation	*14.17*	*605.5*	*1.3742*	*13.50*	*606.2*	*1.3700*	*12.90*	*606.8*	*1.3659*	*12.35*	*607.4*	*1.3621*
−10	14.49	610.3	1.3851	13.74	610.0	1.3784	13.06	609.6	1.3720	12.45	609.2	1.3659
0	14.85	615.9	1.3973	14.09	615.5	1.3907	13.40	615.2	1.3844	12.77	614.8	1.3784
10	15.21	621.3	1.4090	14.44	621.0	1.4025	13.73	620.7	1.3962	13.09	620.4	1.3903
20	15.57	626.7	1.4203	14.78	626.4	1.4138	14.06	626.1	1.4077	13.40	625.8	1.4018
30	15.93	632.0	1.4312	15.11	631.7	1.4248	14.38	631.5	1.4187	13.71	631.2	1.4129
40	16.28	637.3	1.4419	15.45	637.0	1.4356	14.70	636.8	1.4295	14.02	636.6	1.4237
50	16.63	642.5	1.4523	15.78	642.3	1.4460	15.02	642.1	1.4400	14.32	641.9	1.4342
60	16.98	647.7	1.4625	16.12	647.5	1.4562	15.34	647.3	1.4502	14.63	647.1	1.4445
70	17.33	653.0	1.4724	16.45	652.8	1.4662	15.65	652.6	1.4602	14.93	652.4	1.4545
80	17.67	658.2	1.4822	16.78	658.0	1.4760	15.97	657.8	1.4700	15.23	657.7	1.4643
90	18.02	663.4	1.4918	17.10	663.2	1.4856	16.28	663.1	1.4796	15.53	662.9	1.4740
100	18.36	668.6	1.5012	17.43	668.5	1.4950	16.59	668.3	1.4891	15.83	668.1	1.4834
110	18.70	673.8	1.5104	17.76	673.7	1.5042	16.90	673.5	1.4983	16.12	673.1	1.4927
120	19.04	679.1	1.5195	18.08	678.9	1.5133	17.21	678.8	1.5075	16.42	678.6	1.5019
130	19.38	684.3	1.5285	18.41	684.2	1.5223	17.52	684.0	1.5165	16.72	683.9	1.5109
140	19.72	689.5	1.5373	18.73	689.4	1.5312	17.83	689.3	1.5253	17.01	689.2	1.5197
150	20.06	694.8	1.5460	19.05	694.7	1.5399	18.14	694.6	1.5340	17.31	694.4	1.5285
160	20.40	700.1	1.5546	19.37	700.0	1.5485	18.44	699.8	1.5424	17.60	699.7	1.5371
170	20.74	705.4	1.5631	19.70	705.3	1.5569	18.75	705.1	1.5510	17.89	705.0	1.5456
180	21.08	710.7	1.5714	20.62	710.6	1.5653	19.06	710.5	1.5595	18.19	710.4	1.5539
190	21.42	716.0	1.5797	20.31	715.9	1.5736	19.36	715.8	1.5678	18.48	715.7	1.5622
200	21.75	721.3	1.5878	20.66	721.2	1.5817	19.67	721.1	1.5759	18.77	721.1	1.5704
220	22.43	732.1	1.6039	21.30	732.0	1.5978	20.28	731.9	1.5920	19.35	731.8	1.5865

TABLE V.—PROPERTIES OF SUPERHEATED AMMONIA VAPOR (*Continued*)

Absolute pressure in pounds per square inch (saturation temperature in italics)

Temperature, degrees Fahrenheit	23 *−11.25°*			24 *−9.58°*			25			26 *−6.39°*		
								H	S		H	S
Saturation	*11.85*	*608.1*	*1.3584*		*608.6*	*1.3549*	*10.96*	*609.1*	*1.3515*	*10.56*	*609.7*	*1.3482*
−10	*11.89*	*608.8*	*1.3600*									
0	12.20	614.5	1.3726	11.67	614.1	1.3670	11.19	613.8	1.3616	10.74	613.4	1.3564
10	12.50	620.0	1.3846	11.96	619.7	1.3791	11.47		1.3738	11.01	619.1	1.3686
20	12.80	625.5	1.3961	12.25	625.2	1.3907	11.75	625.0	1.3855	11.28	624.	.3804
30	13.10	630.9	1.4073	12.54	630.7	1.4019	12.03		1.3967	11.55	630.	.3917
40	13.40	636.3	1.4181	12.82	636.1	1.4128	12.30	635.8	1.4077	11.81	635.	.4027
50	13.69	641.6	1.4287	13.11	641.4	1.4234	12.57	641.2	1.4183	12.08	641.	.4134
60	13.98	646.9	1.4390	13.39	646.7	1.4337	12.84	646.5	1.	.34	646.	.4238
70	14.27	652.2	1.4491	13.66	652.0	1.4438	13.11	651.8	1.	.59	651.	.4339
80	14.56	657.5	1.4589	13.94	657.3	1.4537	13.37	657.1	1.4487	12.85	656.	.4439
90	14.84	662.7	1.4686	14.22	2.6	1.4634	13.64	2.4	1.4584	13.11	662.	.4536
100	15.13	668.0	1.4780	14.49	667.8	1.4729	13.90	667.7	1.4679	13.36	667.	.4631
110	15.41	673.2	1.4873	14.76	673.1	1.4822	14.17	673.0		13.61	672.	.4725
120	15.70	678.5	1.4965	15.04	678.4	1.4914	14.43	678.2	4864	13.87	678.1	1.4817
130	15.98	638.8	1.5055	15.31	683.6	1.5004	14.69	683.5	1.4954	14.12	83.4	1.4907
140	16.26	689.0	1.5144	15.58	688.9	1.5093	14.95	.8	1.5043	14.37	.7	1.4996
150	16.55	694.3	1.5231	15.85	694.2	1.5180	15.21	694.1	1.5131	14.62	694.	.5084
160	16.83	699.6	1.5317	16.12	699.5	1.5266	15.47	699.4	1.5217	14.87	699.	.5170
170	17.11	704.9	1.5402	16.39	704.8	1.5352	15.73	704.7	1.	.12	704.6	1.5256
180	17.39	710.3	1.5486	16.66	710.2	1.5436	15.99	710.1	1.5387	15.37	710.	.5340
190	17.67	715.6	1.5569	16.93	715.5	1.5518	16.25	715.4	1.5470	15.62	715.	.5423
200	17.95	721.0	1.5651	17.20	720.9	1.5600	16.50	720.8	1.5552	15	720.7	1.5505
220	18.51	731.7	1.5812	17.73	731.7	1.5761	17.02	731.6	1.5713	16.36	731.5	1.5666
240	19.07	742.6	1.5969	18.27	742.6	1.5919	17.53	742.5	1.5870	16.85	742.4	1.5824

Temperature, degrees Fahrenheit	27			28 *−3.40°*			30 *−0.57°*			32 *+2.11°*		
Saturation	*10.20*	*610.2*	*1.3451*	*9.853*	*610.7*	*1.3421*	*9.236*	*611.6*	*1.3364*	*8.693*	*612.4*	*1.3310*
0	10.33	613.0	1.3513	9.942	612.7	1.3465	9.250	611.9	1.3371			
10	10.59	618.8	1.3637	10.20	618.4	1.3589	9.492	617.8	1.3497	8.874	617	1.3411
20	10.85	624.4	1.3755	10.45	624.1	1.3708	9.731	623.5	1.3618	9.099	622	1.3532
30	11.11	629.9	1.3869	10.70	629.6	1.3822	9.966	629.1	1.3733	9.321	628	1.3649
40	11.37	635.4	1.3979	10.95	635.1	1.3933	10.20	634.6	1.3845	9.540	634	1.3762
50	11.62	640.8	1.4087	11.19	640.5	1.4041	10.43	640.1	1.3953	9.757	639.6	1.3871
60	11.87	646.1	1.4191	11.44	645.9	1.4145	10.65	645.5	1.4059	9.972	645.1	1.3977
70	12.12	651.5	1.4292	11.68	651.2	1.4247	10	650.9	1.4161	10.18	650.5	1.4080
80	12.37	656.8	1.4392	11.92	656.6	1.4347	11.10	656.2	.4261	10.40	655.9	1.4181
90	12.61	662.1	1.4489	12.15	661.9	1.4445	11.33	661.6	.4359	10.61	661.2	1.4280
100	12.86	667.4	1.4585	12.39	667.2	.4540	11.55	.9	1.4456	10.81	666.6	1.4376
110	13.10	672.7	1.4679	12.63	672.5	.4634	11.77	672.2	1.4550	11.02	671.9	1.4470
120	13.34	678.0	1.4771	12.86	677.8	.4726	*11.99*	677.	.4642	*11.23*	677.3	*1.4563*
130	13.59	683.3	1.4861	13.10	683.1	1.4817	12.21	682.9	1.4733	11.44	682.6	1.4655
140	13.83	688.6	1.4950	13.33	3.4	1.4906	12.43	2	1.4823	11.64	687.9	1.4744
150	14 07	693.9	1.5038	13.56	693.7	1.4994	12.65	693.5	1.4911	11.85	693.3	1.4833
160	14.31	699.2	1.5125	13.80	9.1	1.5081	12.87		1.4998	12.05	698.6	1.4920
170	14.55	704.5	1.5210	14.03	704.4	1.5167	13.08	704.2	1.5083	12.26	704.0	1.5006
180	14.79	709.9	1.5295	14.26	709.8	1.5251	13.30	709.6	1.5168	12.46	709.4	1.5090
190	15.03	715.2	1.5378	14.49	715.1	1.5334	13.52	714.9	1.5251	12.66	714.7	1.5174
200	15.27	720.6	1.5460	14.72	720.5	1.5416	13.73	720.3	1.5334	12.86	720.1	1.5256
220	15.75	731.4	1.5621	15.18	731.3	1.5578	14.16	731.1	1.5495	13.27	731.0	1.5418
240	16.23	742.3	1.5779	15.64	742.2	1.5736	14.59	742.0	1.5653	13.67	741.9	1.5576
260	16.70	753.2	1.5933	16.10	753.2	1.5890	15.02	753.0	1.5808	14.08	[illegible]	1.5731

TABLE V.—PROPERTIES OF SUPERHEATED AMMONIA VAPOR (*Continued*)

Temperature, degrees Fahrenheit	Absolute pressure in pounds per square inch (saturation temperature in italics)											
	34 *4.66°*			36 *7.09°*			38 *9.42°*			40 *11.66°*		
	V	H	S	V	H	S	V	H	S	V	H	S
Saturation	*8.211*	*613.2*	*1.3260*	*7.782*	*614.0*	*1.3213*	*7.396*	*614.7*	*1.3168*	*7.047*	*615.4*	*1.3125*
10	8.328	616.4	1.3328	7.842	615.7	1.3250	7.407	615.0	1.3175			
20	8.542	622.3	1.3452	8.046	621.7	1.3375	7.603	621.0	1.3301	7.203	620.4	1.3231
30	8.753	628.0	1.3570	8.247	627.4	1.3494	7.795	626.9	1.3422	7.387	626.3	1.3353
40	8.960	633.6	1.3684	8.445	633.1	1.3609	7.983	632.6	1.3538	7.568	632.1	1.3470
50	9.166	639.2	1.3793	8.640	638.7	1.3720	8.170	638.3	1.3650	7.746	637.8	1.3583
60	9.369	644.7	1.3900	8.833	644.2	1.3827	8.353	643.8	1.3758	7.922	643.4	1.3692
70	9.570	650.1	1.4004	9.024	649.7	1.3932	8.535	649.3	1.3863	8.096	648.9	1.3797
80	9.770	655.5	1.4105	9.214	655.2	1.4033	8.716	654.8	1.3965	8.268	654.4	1.3900
90	9.969	660.9	1.4204	9.402	660.6	1.4133	8.895	660.2	1.4065	8.439	659.9	1.4000
100	10.17	666.3	1.4301	9.589	666.0	1.4230	9.073	665.6	1.4163	8.609	665.3	1.4098
110	10.36	671.6	1.4396	9.775	671.3	1.4325	9.250	671.0	1.4258	8.777	670.7	1.4194
120	10.56	677.0	1.4489	9.961	676.7	1.4419	9.426	676.4	1.4352	8.945	676.1	1.4288
130	10.75	682.3	1.4581	10.15	682.1	1.4510	9.602	681.8	1.4444	9.112	681.5	1.4381
140	10.95	687.7	1.4671	10.33	687.4	1.4601	9.776	687.2	1.4534	9.278	686.9	1.4471
150	11.14	693.0	1.4759	10.51	692.8	1.4689	9.950	692.6	1.4623	9.444	692.3	1.4561
160	11.33	698.4	1.4846	10.69	698.2	1.4777	10.12	698.0	1.4711	9.609	697.7	1.4648
170	11.53	703.8	1.4932	10.88	703.6	1.4863	10.30	703.3	1.4797	9.774	703.1	1.4735
180	11.72	709.2	1.5017	11.06	709.0	1.4948	10.47	708.7	1.4883	9.938	708.5	1.4820
190	11.91	714.5	[illegible]	11.21	714.4	1.5032	10.64	714.2	1.4966	10.10	714.0	1.4904
200	12.10	720.0	1.5183	11.42	719.8	1.5115	10.81	719.6	1.5049	10.27	719.4	1.4987
220	12.48	730.8	1.5346	11.78	730.6	1.5277	11.16	730.5	1.5212	10.59	730.3	1.5150
240	12.86	741.7	1.5504	12.14	741.6	1.5436	11.50	741.4	1.5371	10.92	741.3	1.5309
260	13.24	752.7	1.5659	12.50	752.6	1.5591	11.84	752.4	1.5526	11.24	752.3	1.5465
280	13.62	763.8	[illegible]	12.86	763.7	1.5743	12.18	763.5	1.5678	11.55	763.4	1.5617

Temperature, degrees Fahrenheit	42 *13.81°*			44 *15.88°*			46 *17.87°*			48 *19.80°*		
	V	H	S	V	H	S	V	H	S	V	H	S
Saturation	*6.731*	*616.0*	*1.3084*	*6.442*	*616.6*	*1.3046*	*6.177*	*617.2*	*1.3009*	*5.934*	*617.7*	*1.2973*
20	6.842	619.8	1.3164	6.513	619.1	1.3099	6.213	618.5	1.3036	5.937	617.8	1.2976
30	7.019	625.8	1.3287	6.683	625.2	1.3224	6.377	624.6	1.3162	6.096	624.0	1.3103
40	7.192	631.6	1.3405	6.850	631.1	1.3343	6.538	630.5	1.3283	6.251	630.0	1.3225
50	7.363	637.3	1.3519	7.014	636.8	1.3457	6.696	636.4	1.3398	6.404	635.9	1.3341
60	7.531	643.0	1.3628	7.176	642.5	1.3567	6.851	642.1	1.3509	6.554	641.6	1.3453
70	7.697	648.5	1.3734	7.336	648.1	1.3674	7.005	647.7	1.3617	6.702	647.3	1.3561
80	7.862	654.1	1.3838	7.494	653.7	1.3778	7.157	653.3	1.3721	6.848	652.9	1.3666
90	8.026	659.5	1.3939	7.650	659.2	1.3880	7.308	658.9	1.3823	6.993	658.5	1.3768
100	8.188	665.0	1.4037	7.806	664.7	1.3978	7.457	664.4	1.3922	7.137	664.0	1.3868
110	8.349	670.4	1.4133	7.960	670.1	1.4075	7.605	669.8	1.4019	7.280	669.5	1.3965
120	8.510	675.9	1.4228	[illegible]	675.9	1.4170	7.753	675.3	1.4114	7.421	675.0	1.4061
130	8.669	681.3	1.4320	8.267	681.0	1.4263	7.899	680.7	1.4207	7.562	680.5	1.4154
140	8.828	686.7	1.4411	8.419	686.4	1.4351	8.045	686.2	1.4299	7.702	685.9	1.4246
150	8.986	692.1	1.4501	8.570	691.9	1.4444	8.196	691.6	1.4389	7.842	691.4	1.4336
160	9.144	697.5	1.4589	8.721	697.3	1.4532	8.335	697.1	1.4477	7.981	696.8	1.4425
170	9.301	702.9	1.4676	8.871	702.7	1.4619	8.479	702.5	1.4564	8.119	702.3	1.4512
180	9.458	708.3	1.4761	9.021	708.1	1.4704	8.623	707.9	1.4650	8.257	707.7	1.4598
190	9.614	713.8	1.4845	9.171	713.6	1.4789	8.766	713.4	1.4735	8.395	713.2	1.4683
200	9.770	719.2	1.4928	9.320	719.0	1.4872	8.909	718.8	1.4818	8.532	718.7	1.4766
210	9.925	724.7	1.5009	9.474	724.5	1.4954	9.052	724.3	1.4900	8.669	724.2	1.4848
220	10.08	[illegible]	1.5091	9.617	730.0	1.5035	9.194	729.8	1.4981	8.805	729.6	1.4930
240	10.39	[illegible]	1.5251	9.913	741.0	1.5195	9.477	740.8	1.5141	9.077	740.6	1.5090
260	10.70	752.2	1.5406	10.21	752.0	1.5350	9.760	751.9	1.5297	9.348	751.7	1.5246
280	11.01	763.3	1.5559	10.50	763.1	1.5503	10.04	763.0	1.5450	9.619	762.9	1.5399

TABLE V.—PROPERTIES OF SUPERHEATED AMMONIA VAPOR (*Continued*)

Absolute pressure in pounds per square inch (saturation temperature in italics)

Temperature, degrees Fahrenheit	50 *21.67°*			55 *26.09°*			60 *30.21°*			65 *34.06°*		
					H						H	
Saturation	*5.71*	*618.2*	*1.2*	*5.21*	*9.5*		*4.808*		*1.278*	*4.454*	*621.*	*1.2720*
30	5.838	623.4	1.304	5.27	621.9	291						
40	5.988	629.	1.316	5.41	628.1		4.933	626.8	1.291	4.527	625.4	1.2798
50	6.135	635.4	1.328	5.55	634.1	315	5.060		1.303	4.647	631.7	1.2922
60	6.280	641.2	1.339	5.685	640.1	1.327	5.184	639.0	1.315	4.764	637.8	1.3041
70	6.423	646.9	1.350	5.816	645.9	1.338	5.307	644.9	1.326	4.87	643.	1.3156
80	6.564	652.6	1.361	5.947	651.6	1.348	5.428	650.7	1.337	4	649.7	1.3266
90	6.704	658.2	1.371	6.075	657.3	1.359	5.547	656.	1.3479	5.103	655.5	1.3373
100	6.843	663.	1.381	6.202	662.7	1.3694	5.665	662.1	1.3581	5.213	661.3	1.3476
110	6.980	669.2	1.3914	6.329	668.5	1.379	5.781	667.7	1.3681	5.321	667.0	1.3577
120	7.117	674.7	1.400	6.454	674.1	1.3889	5.897	673.3	1.3778	5.429	672.6	1.3675
130	7.25	680.2	1.4103	6.528	679.6	1.3984	6.01	678.9	1.3873	5.536	678.	1.3771
140	7.38	685.	1.4195	6.702	685.1	1.4076	6.126	684.4	1.3966	5.642	683.8	1.3866
150	7.521	691.1	1.4286	6.825	690.6	1.4167	6.239	689.9	1.4058	5.747	689.4	1.3958
160	.655	696.6	1.4374	6.947	696.0	1.4257	6.35	695.5	1.4148	5.852	694.9	1.4048
170	.788	'02.1	1.4462	7.069	701.5	1.4345	6.464	701.0	1.4236	5.956	700.4	1.4137
180	.921	'07.	1.4548	'.190	707.0	1.4431	6.576	706.5	1.4323	6.060	706.0	1.4224
190	8.053	'13.0	1.4633	'.311	'12.5	1.451	6.687	712.0	1.4409	6.163	'11.5	1.4310
200	8.185	'18.5	1.4716	'.43	'18.0	1.4600	6.798	'17.	1.4493	6.266	'17.1	1.4394
210	8.317	'24.0	1.4799	'.552	'23.5	1.4683	6.909	'23.1	1.4576	6.368	'22.6	1.4478
220	8.448	'29.4	.4880	'.671	'29.0	1.4765	.019	'28.6	1.4658	6.471	'28.2	1.4560
240	3.710	'40.5	.5040	'.910	'41.1	1.4925	7.238	'39.7	1.4819	.674	'39.3	1.4722
260	3.970	'51.6	.5197	8.148	'51.2	.5082	.457	'50.9	.4976	6.877	'50.5	.4880
280	.230	'62.7	.5350	8.385	'62.4	.5235	.675	'62.1	.5130	.078	'61.8	.5034
300	9.489	774.0	.5500	3.621	'73.6	.5386	.892	773.3	.5281	.279	'73.1	.5185

Temperature, degrees Fahrenheit	70 *37.70°*			75 *41.13°*			80 *44.40°*			85 *47.50°*		
Saturation	*4.151*		*1.*	*3.88*	*623.2*	*1.259*	*3.658*		*1.2545*	*3.449*	*624.7*	
40	4.177	623.9	1.268									
50	4.290	630	1.281	3.982	629.1	1.2715	3.712	627.7	1.2619	3.473	626.4	1.2527
60	4.401		1.293	4.087	635.5	1.2839	3.812	634.3	1.2745	3.569	633.0	1.2656
70	4.509	642.7	1.305	4.189	641.7	1.295	3.909	640.6	1.2866	3.662	639.5	1.2779
80	4.615	648.7	1.316	4.289	647.7	1.3071	4.005	646.7	1.2981	3.753	645.7	1.2896
90	4.719	654	1.3274	4.388	653.7	1.3180	4.098	652.8	1.3092	3.842	651.8	1.3008
100	4.82	660.4	1.3378	4.485	659.6	1.3286	4.190	658.	1.3199	3.930	657.8	1.3117
110	4.924	666.1	1.3480	4.581	665.4	1.3389	4.281	664.6	1.3303	4.016	663.8	1.3221
120	5.02	671.8	1.3579	4.676	671.1	1.3489	4.371	670.4	1.3404	4.101	669.6	1.3323
130	5.125	677.5	1.3676	4.770	676.8	1.3586	4.460	676.1	1.3502	4.186	675.4	1.3422
140	5.224	683.1	1.3770	4.	682.5	1.3682	4.548	681.8	1.3598	4.269	681.	1.3519
150	5.323	688.7	1.3863	4.956	688.1	1.3775	4.635	687.5	1.369	4.352	686.9	1.3614
160	5.420	694.	.3954	048	693.7	1.3866	4.722	693.2	1.3784	4.434	692.6	1.3706
170	5.518	699.9	.4043	139	699.3	1.3956	4.808	698.8	1.3874	4.51	698.	1.3797
180	5.61	'05.5	.4131	.230	'04.9	1.4044	4.893	704.4	1.3963	4.596	703.9	1.3886
190	.711	'11.0	.4217	.320	'10.5	1.4131	4.978	'10.0	1.4050	4.677	'09.5	1.3974
200	.807	'16.6	1.4302	.410	16.1	.4217	5.063	'15.6	1.4136	4.75'	'15	1.4060
210	.902	'22.	.4386	.500	721.7	.4301	5.147	'21.3	1.4220	4.836	'20.8	1.4145
220	.998	'27.7	.4469	.589	'27.3	.4384	.231	'26.9	1.4304	4.916	'26.4	1.4228
230	6.093	733.3	.4550	.678	'32.9	.4466	.315	732.5	1.4386	.995	'32.1	1.4311
240	.187	'38.9	.4631	.767	738.5	.4546	.398	'38.1	1.4467	.074	'37.7	1.4392
260	.376	'50.1	.4789	.943	'49.8	.4705	.565	'49.4	.4626	.230	'49.0	1.4551
280	.563	'61.4	.4943	.119	'61.1	.4860	.730	'60.7	.4781	.386	'60.4	1.4707
300	.750	'72.7	.5095	.294	'72.4	.5011	.894	772.1	.4933	.541	'71.8	1.4859

TABLE V.—PROPERTIES OF SUPERHEATED AMMONIA VAPOR (*Continued*)

Absolute pressure in pounds per square inch (saturation temperature in italics)

Temperature, degrees Fahrenheit	90 *50.47°*			95 *53.32°*			100 *56.05°* H			105 *58.67°*		
Saturation	*3.266*	*625.3*	*1.2445*	*3.101*	*625.9*	*1.2399*	*2.952*	*626.5*	*1.2356*	*2.817*	*627.0*	*1.2314*
60	3.353	631.8	2571	160	630.5	1.2489	2.985	629.3	1.2409			
70	3.442	638.3	2695	245	637.2	1.2616	3.068	636.0	1.2539	2.907	634.9	1.2464
80	3.529	644.7	1.2814	329	643.6	1.2736	3.149	642.6	1.2661	2.985	641.5	1.2589
90	3.614	650.9	1.2928	3.411	649.9	1.285[illegible]	3.227	649.0	1.2778	3.061	648.0	1.2708
100	3.698	657.0	1.3038	3.491	656.1	1.296	3.304	655.2	1.2891	3.135	654	.2822
110	3.780	663.0	1.3144	3.570	662.1	1.3070	3.380	661.3	1.2999	3.208	660	.2931
120	3.862	668.9	1.3247	3.647	668.1	1.3174	3.454	667.3	1.3104	3.279	666.6	.3037
130	3.942	674.7	1.3347	3.724	674.0	1.3275	3.527	673.3	1.3206	3.350	672.6	.3139
140	4.021	680.5	1.3444	3.799	679.8	1.337[illegible]	3.600	679.2	1.3305	3.419	678.5	1.3239
150	4.100	686.3	1.3539	3.874	685.6	1.3469	3.672	685.0	1.3401	3.488	684.4	1.3336
160	4.178	692.0	1.3633	3.949	691.4	1.3562	3.743	690.8	1.3495	3.556	690.2	1.3431
170	4.255	697.7	1.3724	4.022	697.1	1.3654	3.813	696.6	1.3588	3.623	696.0	1.3524
180	4.332	703.4	1.3813	4.096	702.8	1.3744	3.883	702.3	1.3678	3.690	701.8	1.3615
190	4.408	709.0	1.3901	4.168	708.5	1.3833	3.952	708.0	1.3767	3.757	707.5	1.3704
200	4.484	714.7	1.3988	4.241	714.2	1.3919	4.021	713.7	1.3854	3.823	713.3	1.3792
210	4.560	720.4	1.4073	4.313	719.9	1.400.	4.090	719	3940	3.888	719.0	1.3878
220	4.635	726.0	1.4157	4.384	725.6	1.4089	4.158	725	4024	3.954	724.7	1.3963
230	4.710	731.7	1.4239	4.455	731.3	1.417[illegible]	4.226	730.8	1.4108	4.019	730.4	1.4046
240	4.785	737.3	1.4321	4.526	736.9	1.4254	4.294	736.5	1.4190	4.083	736.1	1.4129
250	4.859	743.0	1.4401	4.597	742.6	1.4334	4.361	742.2	1.4271	4.148	741.9	1.4210
260	4.933	748.7	1.4481	4.668	748.3	1.4414	4.428	747.9	1.4350	4.212	747.6	1.4290
280	5.081	760.0	1.4637	4.808	759.7	.4570	4.562	759.4	1.4507	4.340	759.0	1.4447
290	5.155	765.8	1.4713	4.878	765.5	1.46[illegible]	4.629	765.1	1.4584	4.403	764.8	1.4524
300	5.228	771.5	1.4789	4.947	771	1.4723	4.695	770.8	1.4660	4.466	770.5	1.4600

Temperature, degrees Fahrenheit	110 *61.21°*			115 *63.65°*			120 *66.02°*			125 *68.31°*		
Saturation	*2.693*	*627.5*	*1.2275*	*2.580*	*628.0*	*1.2237*	*2.476*	*628.4*	*1.2201*	*380*	*628.8*	*1.2166*
70	2.761	633.7	1.2392	2.628	632.5	1.2323	2.505	631.3	1.2255	2.392	630.0	1.2189
80	2.837	640.5	1.2519	2.701	639.4	1.2451	2.576	638.3	1.2386	2.461	637.2	1.2322
90	2.910	647.0	1.2640	2.772	646.0	1.2574	2.645	645.0	1.2510	2.528	644.0	1.2448
100	2.981	653.4	1.2755	2.841	652.5	2690	2.712	651.6	1.2628	2.593	650.7	1.2568
110	3.051	659.7	1.2866	2.909	658.8		778	658.	.2741	2.657	657.1	1.2682
120	3.120	665.8	1.2972	2.975	665.0	2910	2.842	664.2	1.2850	2.719	663.5	1.2792
130	3.188	671.9	1.3076	3.040	671.1	1.3015	2.905	670.4	1.2956	2.780	669.7	1.2899
140	3.255	677.8	1.3175	3.105	677.2	1.3116	2.967	676.5	1.3058	2.840	675.8	1.3002
150	3.321	683.7	1.3274	3.168	683.1	1.3215	3.029	682.5	1.3157	2.900	681.8	1.3102
160	3.[illegible]	689.6	1.3370	3.231	689.0	1.3311	3.089	688.4	1.3254	2.958	687.8	1.3199
170	3.451	695.4	1.3463	3.294	694.9	1.3405	3.149	694.3	1.3348	3.016	693.7	1.3294
180	3.515	701.2	1.3555	3.355	700.7	1.3497	3.209	700.2	1.3441	3.074	699.6	1.3387
190	3.579	707.0	1.3644	3.417	706.5	1.3587	3.268	706.0	1.3531	3.131	705.5	1.3478
200	3.642	712.8	1.3732	3.477	712.3	1.3675	3.326	711.8	1.3620	187	711.3	1.3567
210	3.705	718.5	1.3819	3.538	718.1	1.3762	3.385	717.6	1.3707	3.243	717.2	1.3654
220	3.768	724.3	1.3904	3.598	723.8	1.3847	3.442	723.4	1.3793	3.299	723.0	1.3740
230	3.830	730.0	1.3988	3.658	729.6	1.3931	3.500	729.2	1.3877	3.354	728.8	1.3825
240	3.892	735.7	1.4070	3.717	735.3	1.4014	3.557	734.9	1.3960	3.409	734.5	1.3908
250	3.954	741.5	.4151	3.776	741.	1.4096	3.614	740.7	4042	3.464	740.3	1.3990
260	4.015	747.2	1.4232	3.835	746.	1.4176	3.671	746.5	1.4123	3.519	746.1	1.4071
270	4.076	752.9	.4311	3.894	752.	1.4256	3.727	752.2	.4202	3.573	751.9	1.4151
280	4.137	758.7	1.4389	3.952	758.	1.4334	3.783	758.0	.4281	3.627	757.7	1.4230
290	4.198	764.	1 4466	4.011	764.1	1.4411	3.839	763.8	1.4359	3.681	763.5	1.4308
300	4.259	770	1.4543	4.069	769.9	1.4488	3.895	769.6	1.4435	3.735	769.3	1.4385

V.—Properties of Superheated Ammonia Vapor (*Continued*)

Absolute pressure in pounds per square inch (saturation temperature in italics)

Temperature, degrees Fahrenheit	130 *70.53°*			135 *72.69°*			140 *74.79°*			145 *76.83°*		
		H						*H*				
Saturation			*1.2132*	*2.209*	*629.6*	*1.210*			*1.2068*	*2.061*	*630.2*	*1.2038*
80	2.355	636.0	1.2260	2.257	634.9	1.2199	2.166	3.8	.2140	2.080	632.6	1.2082
90	2.421	643.0	1.2388	2.321	642.0	1.2329	2.228	640.9	.2272	2.141	639.9	1.2216
100	2.484	649.7	1.2509	2.382	648.8	1.2452	2.288	647.8	1.2396	2.200	646.9	1.2342
110	2.546	656.3	1.2625	2.44	655.4	1.	.347	654.	2515	2.257	653.6	1.2462
120	2.606	662.7	1.2736	2.501	661.9	1.2681	2.404	661.1	2628	2.313	660.2	1.2577
130	2.6	668.9	1.2843	2.559	668	1.2790	2.460	667.4	2738	2.368	666.	1.2687
140	2.724	675.1	1.2947	2.615	674.4	1.289	2.515	673.7	2843	2.421	673.0	1.2793
150	2.781	681.	1.3048	2.671	680.5	1.2996	2.569	679.9	1.2945	2.474	679.2	1.2896
160	2.838	687.	1.	.726	686.6	1.3094	2.622	686.0	1.3045	2.526	685.4	1.2996
170	2.894	693.	1.3241	2.780	692.6	1.3191	2.675	692.0	1.3141	2.577	691.4	1.3093
180	2.949	699.1	1.3335	2.834	698.6	1.3284	2.727	.0	1.3236	2.627	97.	1.3188
190	3.004	705.0	1.3426	2.887	704.5	1.3376	2.779	704.0	1.3328	2.677	703.4	1.3281
200	3.059	710.9	1.3516	2.940	710.4	1.3466	2.830	709.9	1.3418	2.727	709.4	1.3372
210	3.113	716	1.3604	2.992	716.	1.3554	2.880	715.8	1.3507	2.776	715.3	1.3461
220	3.167	722.5	1.3690	3.044	722	1.3641	2.931	721.6	1.3594	2.825	721.	1.3548
230	3.220	728	1.3775	3.096	727.9	1.3726	2.981	727.5	1.3679	2.873	727.1	1.3634
240	3.273	734	1.3858	3.147	733.7	1.3810	3.030	733.3	1.3763	2.921	'32.9	1.3718
250	3.326	'39.9	1.3941	3.198		1.3893	3.080	739.2	1.3846	2.969	738.8	1.3801
260	3.379	'45.7	1.402	3.249		1.3974	3.129	745.0	1.3928	3.017	'44.6	1.3883
270	3.431	.5	1.4102	3.300		1.4054	3.179	750.8	1.4008	3.064	'50.5	1.3964
280	3.483		1.4181	3.350	.0		3.227	756.7	1.4088	3.111	'56.3	1.4043
290	3.535		1.4259	3.400		1.4212	3.275	'62.5	1.4166	3.158	'62.2	1.4122
300	3.587	'69.0	1.4336	3.450	'68.6	1.4289	3.323		1.4243	3.205	'68.0	.4199
320	3.690	'80.6	1.4487	3.550	'80.3	1.4441	3.420	780.0	1.4395	3.298	'79.7	.4352

Temperature, degrees Fahrenheit	150 *78.81°*			155 *80.75°*	160 *82.64°*			165 *84.49°*
Saturation	*1.994*	*630.*	*1.200*		*1.872*	*631.1*	*1.1952*	
90	2.061	.8	1.216		1.914	636.6	1.2055	
100	2.118	645.9	1.2289		1.969	643.9	2186	
110	2.174	652.8	1.2410		2.023	651.0	2311	
120	2.228	659.4	1.2526		2.075	657.8	2429	
130	2.281	665.9	1.2638		2.12	664.4	2542	
140	2.334	672.3	1.2745		2.175	670.9	1.2652	
150	2.385	678.6	2849			677.	1.2757	
160	2.435	684.8	2949			683.	1.2859	
170	2.485	690.9	304'		2.319	689.7	1.2958	
180	2.534	.9	1.314		2.365	695.8	1.3054	
190	2.583	702.9	1.3236		2.411	701.9	1.3148	
200	2.631	708.9	.332'	(Data not available)	2.457	707.9	3240	(Data not available)
210	2.679	714.8	.3416		2.502	713.9	3331	
220	2.726	720.7	.3504		2.547	719.9	3419	
230	2.773	726.6	.3590		2.591	725.8	3506	
240	2.820	732.5	1.367		2.635	731.7	1.3591	
250		738.4	1.3758		2.679	737.6	1.3675	
260	2.912	'44.3	1.3840		2.723	743.5	1.3757	
270	2.958	750.1	1.3921		2.766	749.4	1.3838	
280	3.004	756.0	1.4001		2.809	755.3	1.3919	
290	3.049	'61.	1.4079		2.852	761.2	1.3998	
300	3.095	'67.7	1.4157			767.1	1.4076	
320	3.185	'79.4	1.4310			778.9	1.4229	
340	3.274	'91.2	1.4459		3.064	790.7	1.4379	

TABLE V.—PROPERTIES OF SUPERHEATED AMMONIA VAPOR (*Continued*)

Absolute pressure in pounds per square inch (saturation temperature in italics)

Temperature, degrees Fahrenheit	170 *86.29°*			180 *89.78°*			190 *93.13°*			200 *96.34°*		
		H			H							
Saturation	*1.*	*.6*	*1.1900*	*1.667*	*632.0*	*1.1850*	*1.581*		*1.1802*	*1.502*	*632.7*	*1.1756*
90	1.784	634.4	1.1952	1.668	632.2	1.1853						
100	1.837	641.9	1.2087	1.720	639.9	1.1992	1.615	637.8	1.1899	1.520	635.	1.1809
110	1.889	649.1	1.2215	1.770	647.3	1.2123	1.663	645.4	1.2034	1.567	643.	1.1947
120	1.939	656.1	1.2336	1.818	654.4	1.2247	1.710	652.6	1.2160	1.612	650.9	1.2077
130	1.988	662.8	1.2452	1.865	661.	.2364	1.755	659.7	1.2281	1.656	658	1.2200
140	2.035	669.1	1.2563	[illegible]	668.	.2477	1.799	666.5	1.2396	1.698	665	1.2317
150	2.081	675.9	1.2669	1.955	674.	1.2586	842	673.2	1.2506	1.740	8	1.2429
160	2.127	682.3	1.2773	1.999	681.0	1.2691	884	679.7	1.2612	1.780	678.4	1.2537
170	2.172	688.5	1.2873	2.042	687.3	1.2792	925	686.1	1.2715	1.820	684.9	1.2641
180	2.216	694.7	1.2971	2.084	693.6	1.2891	966	692.5	1.2815	1.859	691.3	1.2742
190	2.260	700.8	1.3066	2.126	699.8	1.2987	2.005	698.7	1.2912	1.897	697.7	1.2840
200	2.303	706.9	1.3159	2.167	705.9	1.3081	2.045	704.9	1.3007	1.935	703.9	1.2935
210	2.346	713.0	1.3249	2.208	712.0	1.3172	2.084	711.1	1.3099	1.972	710.1	1.3029
220	2.389	719.0	1.3338	2.248	718.1	1.3262	2.123	717.2	1.3189	2.009	716.3	1.3120
230	2.431	724.9	1.3426	2.288	724.1	1.3350	2.161	723.2	1.3278	2.046	722.4	1.3209
240	2.473	730.9	1.3512	2.328	730.1	1.3436	2.199	729.3	1.3365	2.082	728.4	1.3296
250	2.514	736.8	1.3596	2.367	736.1	1.3521	2.236	735.3	1.3450	2.118	734.5	1.3382
260	2.555	742.8	1.3679	2.407	742.0	1.3605	2.274	741.3	1.3534	2.154	740.5	1.3467
270	2.596	748.7	1.3761	2.446	748.0	1.3687	2.311	747.3	1.3617	2.189	746	1.3550
280	2.637	754.6	1.3841	2.484	753.9	1.3768	2.348	753.2	1.3698	2.225	752	1.3631
290	2.678	760.5	1.3921	2.523	759.9	1.3847	2 384	759.2	1.3778	2.260	758.5	1.3712
300	2.718	766.4	3999	2.561	765.8	1.3926	2.421	765.2	1.3857	2.295	764.5	1.3791
320	2.798	778.3	4153	2.637	777.7	1.4081	2.493	777.1	1.4012	2.364	776.5	1.3947
340	2.878	790.1	1.4303	2.713	789.6	1.4231	2.565	789.0	1.4163	2.432	788.5	1.4099

Temperature, degrees Fahrenheit	210 *99.43°*			220			230 *105.30°*			240 *108.09°*		
Saturation	*1.431*	*633.0*	*1.1713*	*1.367*	*633.2*	*1.1671*	*1.307*	*633.4*	*1.1631*	*1.253*	*633.6*	*1.1592*
110	1.480	641	1.1863	1.400	639.4	1.1781	1.328	637.4	1.1700	1.261	635.3	1.1621
120	1.524	649	1.1996	1.443	647.3	1.1917	1.370	645.4	1.1840	1.302	643.5	1.1764
130	1.566	656	1.2121	1.485	654.8	1.2045	1.410	653.1	1.1971	1.342	651.3	1.1898
140	1.608	663	1.2240	1.525	662.0	1.2167	1.449	660.4	1.2095	1.380	658.8	1.2025
150	1.648	670.4	1.2354	1.564	669.0	1.2281	1.487	667.6	1.2213	1.416	666.1	1.2145
160	1.687	677.1	1.2464	1.601	675.8	1.2394	1.524	674.5	1.2325	1.452	673.1	1.2259
170	1.725	683.7	1.2569	1.638	682.5	1.2501	1.559	681.3	1.2434	1.487	680.0	1.2369
180	1.762	690.2	1.2672	1.675	689.1	1.2604	1.594	687.9	1.2538	1.521	.7	1.2475
190	1.799	696.6	1.2771	1.710	695.5	1.2704	1.629	694.4	1.2640	1.554	693.3	1.2577
	1.836	702.9	1.2867	1.745	701.9	1.2801	1.663	700.9	1.2738	1.587	699.8	1.2677
210	1.872	709.2	1.2961	780	708.2	1.2896	1.696	707.2	1.2834	1.619	706.2	1.2773
220	1.907	715.3	1.3053	1.814	714.4	1.2989	1.729	713.5	1.2927	1.651	712.6	1.2867
230	1.942	721.5	1.3143	1.848	720.6	1.3079	1 762	719.8	1.3018	1.683	718.9	1.2959
240	1.977	727.6	1.3231			1.3168	1 794	726.0	1.3107	1.714	725.1	1.3049
250	2.011	733.7	1.331'	914	732.9	.3255	1.826	732.1	1.3195	1.745	731.3	1.3137
260	2.046	739.8	1.340	947	739.0	.3340	1.857	738.3	1.3281	1.775	737.5	1.3224
270	2.080	745.8	1.3486	1.980	745.1	3424	1.889	744.4	1.3365	1.805	743.6	1.3308
280	2.113	751.8	1.3568	2.012	751.1	.3507	1.920	750.5	1.3448	1.835	749.8	1.3392
290	2.147	757.9	1.3649	2.044	757.2	3588	1.951	756.5	1.3530	1.865	755.9	1.3474
300	2.180	763.9	1.3728	2 076	763.2	1.3668	1.982	762.6	1.3610	1.895	762.0	1.3554
320	2.246	775.	1.3884	2.140	775.3	1.3825	2.043	774.7	1.3767	1.954	774.1	1.3712
340	2.312	787.9	1.403'	2.203	787.4	1.3978	2.103	786.8	1.3921	2.012	786.3	1.3866
360	2.377	800.0	1.4186	2.265	799.5	1.4127	2.163	798.9	1.4070	2.069	798.4	1.4016
380	2.442	812.0	1.4331	2.327	811 6	1.4273	2.222	811.1	1.4217	2.126	810.6	1.4163

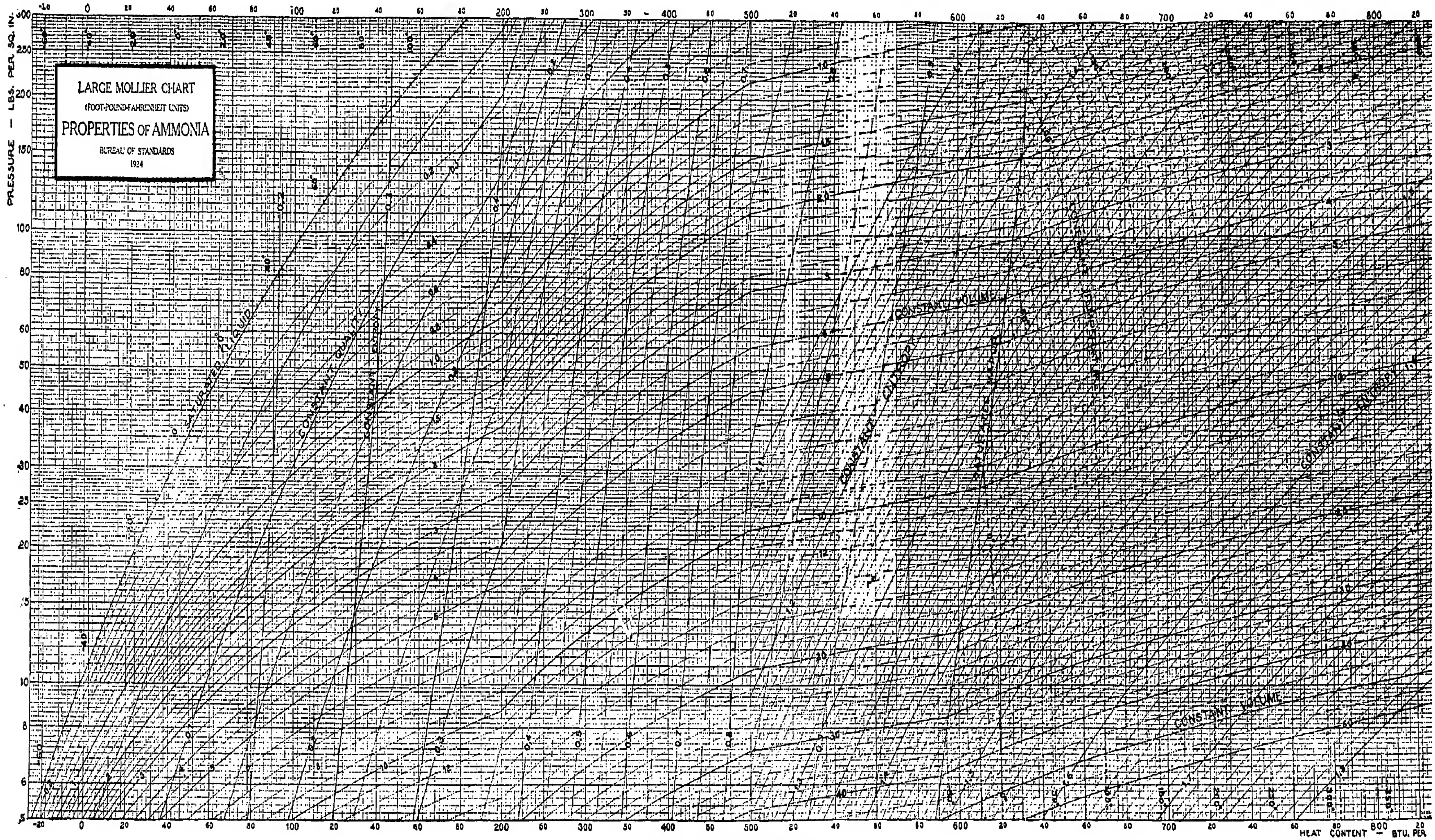

Fig. 292.—Mollier chart of properties of ammonia.

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TABLE VI.—PROPERTIES OF SATURATED DICHLORODIFLUOROMETHANE F-12

Temperature, degrees Fahrenheit t	Pressure: Absolute, pounds per square inch p	Pressure: Gage, pounds, per square inch $g.p.$	Volume vapor, cubic feet per pound V	Density vapor, pounds per cubic foot $1/V$	Heat content: Liquid, B.t.u. per pound h	Heat content: Latent heat, B.t.u. per pound L	Heat content: Vapor, B.t.u. per pound H	Entropy: Liquid, B.t.u., per pound degrees Fahrenheit s	Entropy: Vapor, B.t.u., per pound degrees Fahrenheit S	Temperature, degrees Fahrenheit t
−40	9.32	10.92*	3.911	0.2557	0	73.50	73.50	0	0.17517	**−40**
−38	9.82	9.91*	3.727	0.2683	0.40	73.34	73.74	0.00094	0.17490	−38
−36	10.34	8.87*	3.553	0.2815	0.81	73.17	73.98	0.00188	0.17463	−36
−34	10.87	7.80*	3.389	0.2951	1.21	73.01	74.22	0.00282	0.17438	−34
−32	11.43	6.66*	3.234	0.3092	1.62	72.84	74.46	0.00376	0.17412	−32
−30	12.02	5.45*	3.088	0.3238	2.03	72.67	74.70	0.00471	0.17387	**−30**
−28	12.62	4.23*	2.950	0.3390	2.44	72.50	74.94	0.00565	0.17364	−28
−26	13.26	2.93*	2.820	0.3546	2.85	72.33	75.18	0.00659	0.17340	−26
−24	13.90	1.63*	2.698	0.3706	3.25	72.16	75.41	0.00753	0.17317	−24
−22	14.58	0.24*	2.583	0.3871	3.66	71.98	75.64	0.00846	0.17296	−22
−20	15.28	0.58	2.474	0.4042	4.07	71.80	75.87	0.00940	0.17275	**−20**
−18	16.01	1.31	2.370	0.4219	4.48	71.63	76.11	0.01033	0.17253	−18
−16	16.77	2.07	2.271	0.4403	4.89	71.45	76.34	0.01126	0.17232	−16
−14	17.55	2.85	2.177	0.4593	5.30	71.27	76.57	0.01218	0.17212	−14
−12	18.37	3.67	2.088	0.4789	5.72	71.09	76.81	0.01310	0.17194	−12
−10	19.20	4.50	2.003	0.4993	6.14	70.91	77.05	0.01403	0.17175	**−10**
−8	20.08	5.38	1.922	0.5203	6.57	70.72	77.29	0.01496	0.17158	−8
−6	20.98	6.28	1.845	0.5420	6.99	70.53	77.52	0.01589	0.17140	−6
−4	21.91	7.21	1.772	0.5644	7.41	70.34	77.75	0.01682	0.17123	−4
−2	22.87	8.17	1.703	0.5872	7.83	70.15	77.98	0.01775	0.17107	−2
0	23.87	9.17	1.637	0.6109	8.25	69.96	78.21	0.01869	0.17091	**0**
2	24.89	10.19	1.574	0.6352	8.67	69.77	78.44	0.01961	0.17075	2
4	25.96	11.26	1.514	0.6606	9.10	69.57	78.67	0.02052	0.17060	4
5†	**26.51**	**11.81**	**1.485**	**0.6735**	**9.32**	**69.47**	**78.79**	**0.02097**	**0.17052**	**5**†
6	27.05	12.35	1.457	0.6864	9.53	69.37	78.90	0.02143	0.17045	6
8	28.18	13.48	1.403	0.7129	9.96	69.17	79.13	0.02235	0.17030	8
10	29.35	14.65	1.351	0.7402	10.39	68.97	79.36	0.02328	0.17015	**10**
12	30.56	15.86	1.301	0.7687	10.82	68.77	79.59	0.02419	0.17001	12
14	31.80	17.10	1.253	0.7981	11.26	68.56	79.82	0.02510	0.16987	14
16	33.08	18.38	1.207	0.8288	11.70	68.35	80.05	0.02601	0.16974	16
18	34.40	19.70	1.163	0.8598	12.12	68.15	80.27	0.02692	0.16961	18
20	35.75	21.05	1.121	0.8921	12.55	67.94	80.49	0.02783	0.16949	**20**
22	37.15	22.45	1.081	0.9251	13.00	67.72	80.72	0.02873	0.16938	22
24	38.58	23.88	1.043	0.9588	13.44	67.51	80.95	0.02963	0.16926	24
26	40.07	25.37	1.007	0.9930	13.88	67.29	81.17	0.03053	0.16913	26
28	41.59	26.89	0.973	1.028	14.32	67.07	81.39	0.03143	0.16900	28
30	43.16	28.46	0.939	1.065	14.76	66.85	81.61	0.03233	0.16887	**30**
32	44.77	30.07	0.908	1.102	15.21	66.62	81.83	0.03323	0.16876	32
34	46.42	31.72	0.877	1.140	15.65	66.40	82.05	0.03413	0.16865	34
36	48.13	33.43	0.848	1.180	16.10	66.17	82.27	0.03502	0.16854	36
38	49.88	35.18	0.819	1.221	16.55	65.94	82.49	0.03591	0.16843	38
40	51.68	36.98	0.792	1.263	17.00	65.71	82.71	0.03680	0.16833	**40**
42	53.51	38.81	0.767	1.304	17.46	65.47	82.93	0.03770	0.16823	42
44	55.40	40.70	0.742	1.349	17.91	65.24	83.15	0.03859	0.16813	44
46	57.35	42.65	0.718	1.393	18.36	65.00	83.36	0.03948	0.16803	46
48	59.35	44.65	0.695	1.438	18.82	64.74	83.57	0.04037	0.16794	48
50	61.39	46.69	0.673	1.485	19.27	64.51	83.78	0.04126	0.16785	**50**
52	63.49	48.79	0.652	1.534	19.72	64.27	83.99	0.04215	0.16776	52
54	65.63	50.93	0.632	1.583	20.18	64.02	84.20	0.04304	0.16767	54
56	67.84	53.14	0.612	1.633	20.64	63.77	84.41	0.04392	0.16758	56
58	70.10	55.40	0.593	1.686	21.11	63.51	84.62	0.04480	0.16749	58

* Inches of mercury below one atmosphere.
† Standard ton temperatures.

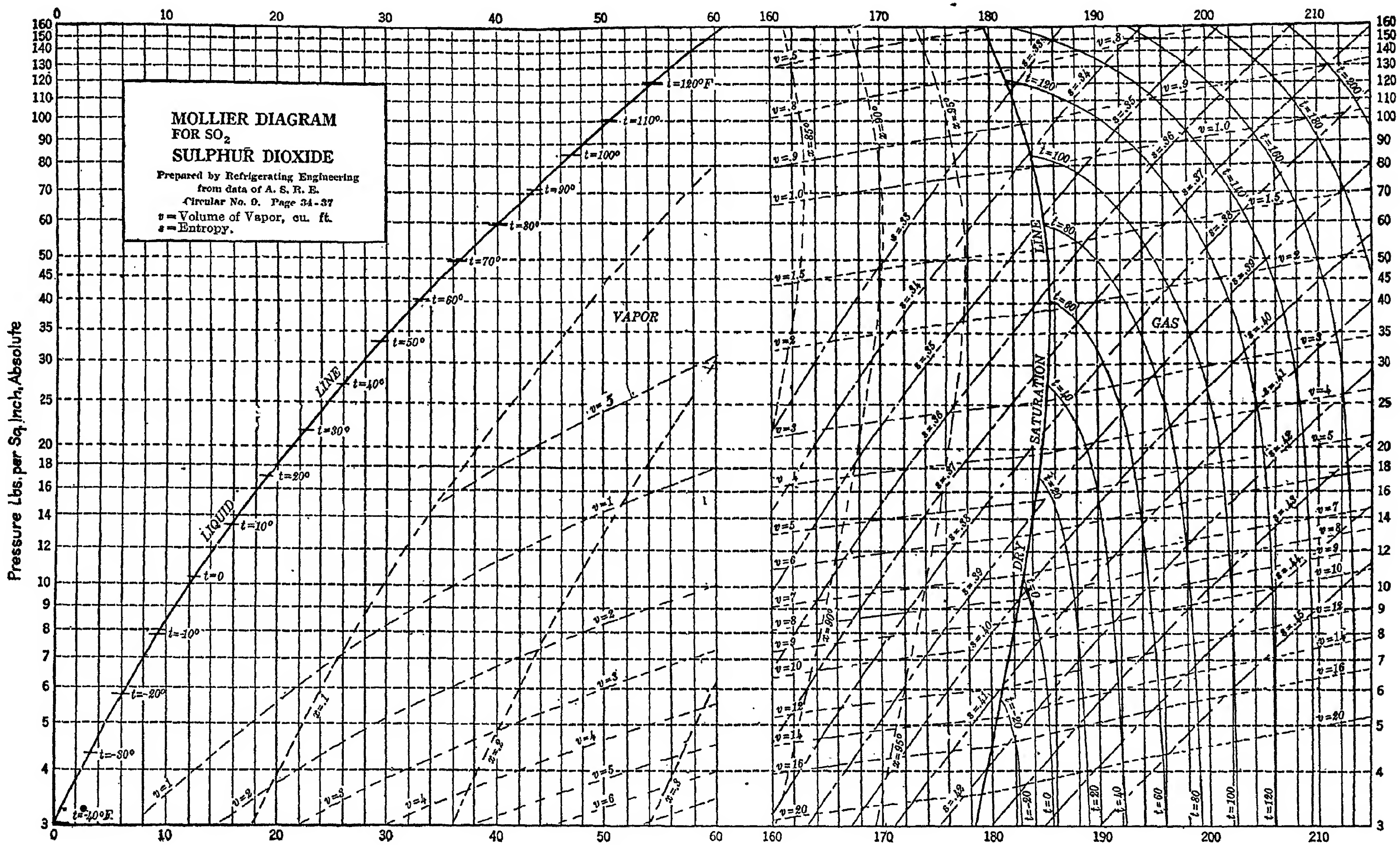

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FIG. 293.—Mollier chart of propreties of sulphur dioxide.

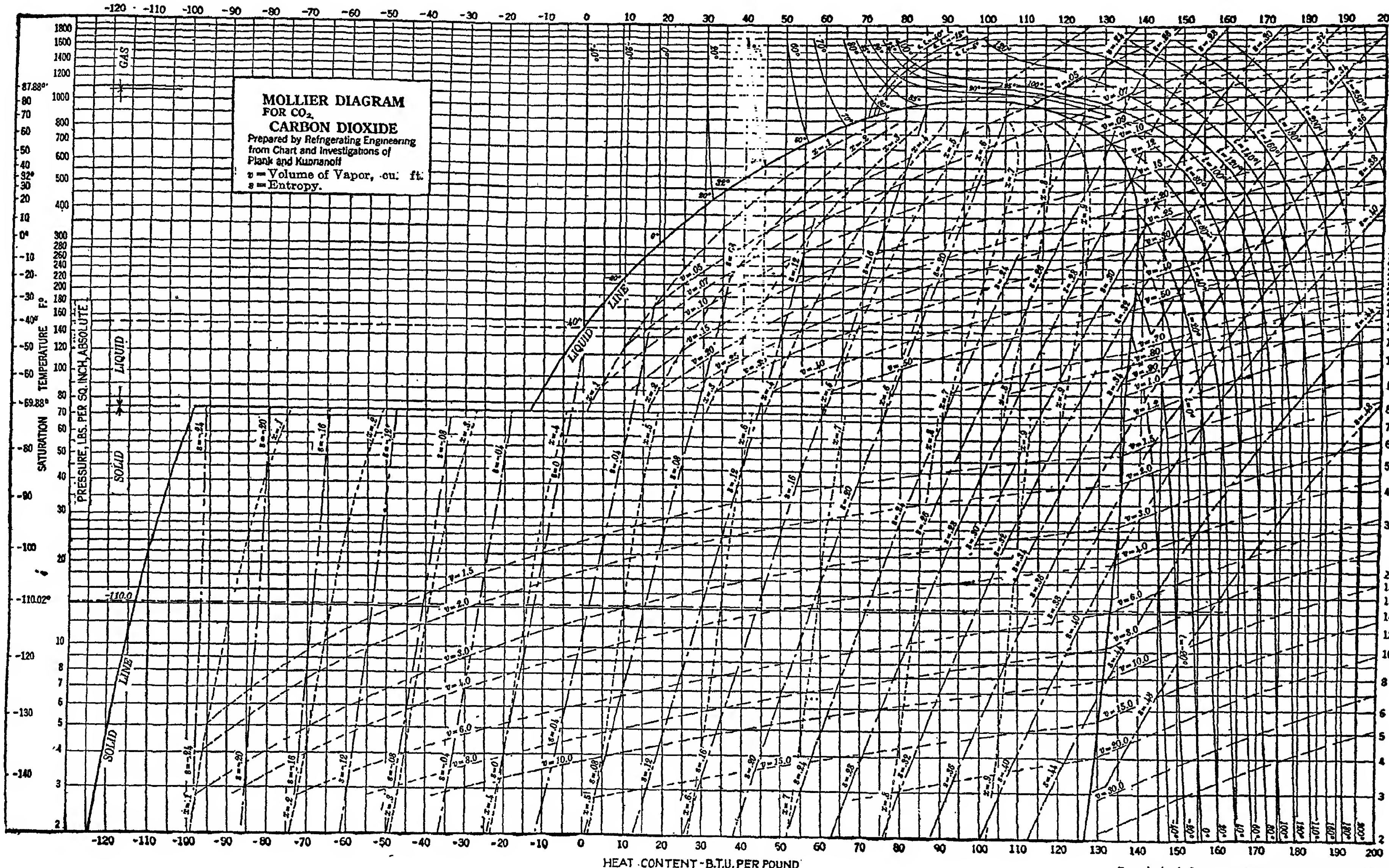

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FIG. 294.—Mollier chart of properties of carbon dioxide.

TABLE VII.—FREEZING TEMPERATURE OF FRUITS AND VEGETABLES*

Fruits	Degrees Fahrenheit
Apple	30.20
Banana	27.32
Cantaloupe	30.74
Grapefruit	29.66
Lemon	31.10
Grape	28.40
Orange	29.84
Peach	31.10
Pear	29.12
Plume	29.84

Vegetables	Degrees Fahrenheit
Bean, green	31.28
Beets	31.10
Cabbage	31.28
Carrot	30.20
Celery	31.10
Corn, green	31.10
Cucumber	31.46
Lettuce	31.82
Onion	30.74
Irish potato	31.64
Sweet potato	29.84
Squash	31.28
Tomato	31.10

* SMITH, AUBREY L., "Freezing and Melting Points of Fruits and Vegetables," *Refrigerating Eng.*, Vol. 21, No. 4.

TABLE VIII.—PROPERTIES OF SATURATED VAPOR OF METHYL CHLORIDE (CH_3Cl)

Temperature, degrees Fahrenheit	Pressure, pounds per square inch absolute	Volume of vapor, cubic feet per pound	Density of vapor, pounds per cubic foot	Heat content above 32° F., B.t.u. per pound		Total latent heat, B.t.u. per pound			Entropy, B.t.u. per pound per degrees Fahrenheit absolute		
				Liquid	Vapor	Total r	Inner r_1 Apu.	Outer Apu.	Liquid	Vapor	Evaporation
−40	6.96	12.57	0.079	−34.0	149.3	183.3	167.1	16.17	−0.075	0.361	0.436
−31	8.84	10.07	0.099	−29.7	152.9	182.6	166.2	16.43	−0.065	0.360	0.425
−22	11.11	8.16	0.123	−25.5	156.2	181.7	165.0	16.70	−0.055	0.359	0.414
−13	13.82	6.66	0.150	−21.2	159.6	180.8	163.8	16.97	−0.045	0.359	0.404
− 4	17.07	5.47	0.183	−17.0	162.9	179.9	162.7	17.20	−0.036	0.358	0.394
+ 5	20.89	4.53	0.221	−12.7	165.8	178.5	161.1	17.44	−0.027	0.357	0.384
14	25.38	3.78	0.265	− 8.5	168.8	177.3	159.7	17.65	−0.018	0.356	0.374
23	30.60	3.17	0.315	− 4.2	171.9	176.1	158.3	17.84	−0.009	0.355	0.364
32	36.63	2.68	0.373	0.0	174.6	174.6	156.6	18.02	0.000	0.354	0.354
41	43.60	2.27	0.440	4.2	177.1	172.9	154.7	18.19	0.009	0.354	0.345
50	51.60	1.94	0.516	8.5	179.6	171.1	152.7	18.43	0.017	0.352	0.335
59	60.50	1.66	0.601	12.7	182.0	169.3	150.8	18.47	0.025	0.351	0.326
68	71.00	1.43	0.698	17.0	184.3	167.3	148.8	18.55	0.033	0.349	0.316
77	82.53	1.24	0.807	21.2	186.4	165.2	146.6	18.62	0.041	0.348	0.307
86	95.53	1.07	0.930	25.5	188.4	162.9	144.2	18.69	0.049	0.347	0.298
95	110.10	0.94	1.067	29.7	190.2	160.5	141.8	18.70	0.057	0.346	0.289
104	126.26	0.82	1.222	34.0	191.8	157.8	139.1	18.70	0.064	0.343	0.279

TABLE IX.—PHYSICAL PROPERTIES OF REFRIGERANTS

Refrigerant	Ammonia	Butane	Carbon dioxide	Ethane	Ethyl chloride	Methyl chloride	Propane	Sulphur dioxide
Chemical symbol	NH_3	C_4H_{10}	CO_2	C_2H_6	C_2H_5Cl	CH_3Cl	C_2H_8	SO_2
Molecular weight	17.032	58.10	44.005	30.058	64.51	50.489	44.079	64.06
Color of liquid	colorless	colorless	colorless	colorless	colorless	colorless	colorless	colorless
Odor	pungent and aromatic	slight, like illuminating gas	odorless	slight, like illuminating gas	pungent and ethereal odor	similar to chloroform	slight, like illuminating gas	pungent
Density of liquid (water = 1)	0.6818	0.60	See Note X	0.5459	0.9232	0.998	0.5853	1.4601
at temperature in degrees Centigrade	−33.35	0.0	....	−88.3	0.0	−24.09	−44.5	−10.0
Density of gas, grams 1 liter (see Note Y)	0.7708	2.6726	1.9768	1.3565	....	2.3045	2.0200	2.9267
(air = 1)	0.5962	2.0671	1.5290	1.0492	2.31	1.7824	1.5624	2.2636
Boiling point at 1 atmosphere, degrees Centigrade	−33.35	0.6	See Note X	−88.3	13.1	−24.09	−44.5	−10.0
at temperature in degrees Fahrenheit	−28.03	33.1	....	−126.9	55.6	−11.36	−48.1	14.0
Melting point, degrees Centigrade	−77.70	−135.0	−78.52	−172.0	−138.7	−91.5	−189.9	−75.2
degrees Fahrenheit	−107.86	−211.0	−109.34	−277.6	−217.7	−132.7	−309.8	−103.4
Critical temperature, degrees Centigrade	132.9	150.8	31.00	32.1	182.8	143.12	95.6	157.12
degrees Fahrenheit	271.2	303.4	87.80	89.8	361.0	289.6	204.1	314.82
Critical pressure, atmosphere absolute	112.3	37.5	72.85	48.85	53.3	65.93	45	77.65
pounds per square inch absolute	1,651	551.3	1,071	718.0	784	969.2	661.5	1,141.5
Specific heat of constant pressure (C_p)	0.5202	0.351	0.2025	0.397	0.273	0.24	0.365	0.1511
Specific heat of constant volume (C_v)	0.4011	....	0.1558	....	....	0.20		
Ratio of specific heats (C_p/C_v)	1.2969	1.108	1.3003	1.224	1.1257	1.1991	1.153	1.256
at degrees Centigrade	....	....	0.0	10	73	66–68	....	16–34
Latent heat of vaporization at 1 atmosphere, B.t.u. per pound	589.4	165.4	256.3 See Note X	225	168.6	180.6	180.4	172.3
Suction pressure at 5° F., pounds per square inch absolute	34.5	8.7	331.8	235	4.65	20.89	43.5	11.82
Head pressure at 86° F., pounds per square inch absolute	168.5	42.7	1,039.6	677	27.1	95.53	159	65.9

NOTE X.—Carbon dioxide not a liquid at atmospheric pressure.
NOTE Y.—Density of gas at 0° C. (32° F.) and 760 millimeters (1 atmosphere), except for ethyl chloride, in which no temperature has been given.
For detailed data and references, see following tables.

TABLE X.—COLD-STORAGE DATA*

Name	Percentage water	Storage temperature		Specific heat		Latent heat of fusion
		Low, degrees Fahrenheit	Normal, degrees Fahrenheit	After freezing	Before freezing	
Apples	83	29	31	....	0.92	
Asparagus	94	33	34			
Bacon, smoked	..	30	32			
Bananas	..	35	40			
Beans, green	89	32	33			
Beans, dried	..	....	40			
Beef, lean	72	30	32	0.41	0.77	102
Beef, fat	51	30	32	0.34	0.60	72
Beef, fresh, chilled	..	30	32			
Beef, freezing	..	5	10			
Beef, frozen	..	15	20			
Beef, storage	..	....	33			
Beef, dried	..	....	40			
Berries	..	31	40	....	0.42	
Butter, tubs	..	0	15	....	0.55	
Butter, cartons	..	0	15			
Cabbage	91	25	31	0.43	0.93	129
Cantaloupes	..	33	36			
Carrots	83	30	36	0.45	0.87	113
Cauliflower	93	22				
Celery	95	10	33			
Cheese, cream	..	30	32	....	0.64	
Cheese, brick	..	30	32			
Cherries, fresh	82	....	40			
Cider	..	30	32			
Corn, green	75	....	38			
Cranberries	..	28	33			
Cream, fresh	59	32	34	0.38	0.90	84
Cucumbers	95	32	38			
Eggs, freezing	70	−10	0	0.40	0.76	100
Eggs, storage	70	28	29	0.40	0.76	100
Fruits, dried	..	40				
Fruits, canned	..	40				
Fish, dried	..	35				
Fish, freezing	73	−15	..	0.43	0.82	111
Fish, storage	..	....	26			
Fish, frozen	..	5	18			
Flour and meal	..	....	40			
Furs	..	25	35			
Furs, undressed	..	....	35			
Flowers, cut	..	....	36			
Game, freezing	..	5	10			
Game, storage	..	15	25			
Grapes, fresh	..	26	32			
Grapefruit	..	32	36			
Ice cream	67	0	15	0.45	0.78	90

* From tables prepared by W. H. Motz.

TABLE X.—COLD-STORAGE DATA (*Continued*)

Name	Percentage water	Storage temperature: Low, Fahrenheit	Storage temperature: Normal, degrees Fahrenheit	Specific heat: After freezing	Specific heat: Before freezing	Latent heat of fusion
Lard		32	38	0.31	0.54	
Lemons		36	38			
Lettuce	94	26	42			
Lobsters	77	...	25	0.42	0.81	108
Melons		33	40			
Milk, fresh		32	36	0.47	0.90	124
Mutton, chilling		30	32	0.67	0.81	100
Onions	88	32	35			
Oranges		32	35			
Oysters, shell	80	30	35	0.44	0.84	114
Parsnips	83	32	33			
Peaches, fresh	87	30	30			
Pears, fresh	83	30	32			
Peas, fresh	75	32	36			
Pineapple		32	40			
Plums, fresh		28	32			
Pork, salt			42			
Pork, chill	39	30	32	0.30	0.51	55
Pork, storage			32			
Pork, freeze		5	10			
Potatoes	73	30	33			
Potatoes, sweet	69	50	55			
Poultry, freeze	74	0	10	0.42	0.80	105
Poultry, storage		28	30			
Poultry, frozen		10	15		0.377	
Strawberries	90	33	40			
Tomatoes	94	33	34			
Veal	63		34	0.39	0.70	90

Table XI.—Properties of Calcium-chloride Brine

U. S. Bureau of Standards

Temperature, degrees Fahrenheit	Specific gravity									
	0.999	1.05	1.10	1.15	1.18	1.20	1.22	1.24	1.26	1.28
	Pounds per gallon									
70	8.33	8.75	9.17	9.58	9.83	10.00	10.16	10.33	10.49	10.66
60	8.34	8.76	9.18	9.60	9.85	10.01	10.18	10.35	10.52	10.68
50	8.34	8.77	9.19	9.61	9.86	10.03	10.20	10.37	10.54	10.70
40	8.35	8.78	9.21	9.63	9.88	10.05	10.22	10.39	10.56	10.73
30		8.79	9.22	9.64	9.90	10.07	10.24	10.41	10.58	10.75
20			9.23	9.66	9.92	10.09	10.26	10.43	10.60	10.77
10				9.68	9.93	10.11	10.28	10.45	10.62	10.79
0					9.95	10.13	10.30	10.47	10.64	10.81
−10							10.32	10.49	10.66	10.84
−20									10.68	10.86
−30									10.70	10.88
	Pounds per cubic foot									
70	62.3	65.5	68.6	71.7	73.5	74.8	76.0	77.3	78.5	79.7
60	62.4	65.6	68.7	71.8	73.7	74.9	76.2	77.4	78.7	79.9
50	62.4	65.6	68.8	71.9	73.8	75.0	76.3	77.6	78.8	80.1
40	62.4	65.7	68.9	72.0	73.9	75.2	76.4	77.7	79.0	80.3
30	......	64.8	69.0	72.1	74.0	75.3	76.6	77.9	79.2	80.4
20	......	.....	69.1	72.3	74.2	75.4	76.7	78.0	79.3	80.6
10	......	.....	.....	72.4	74.3	75.6	76.9	78.2	79.5	80.8
0	......	.....	.....	.....	74.4	75.7	77.0	78.3	79.6	80.9
−10	......	.....	.....	.....	.....	.....	77.2	78.5	79.8	81.1
−20	......	.....	.....	.....	.....	.....	.....	.....	80.0	81.3
−30	......	.....	.....	.....	.....	.....	.....	.....	80.1	81.4

Table XII.—Specific Heats of Calcium-chloride Solutions*

Temperature, degrees Fahrenheit	Densities (pounds per cubic foot)			
	1.175	1.200	1.225	1.250
−10	.....	.....	0.670	0.654
0	0.722	0.697	0.676	0.659
+10	0.728	0.703	0.681	0.663
+20	0.733	0.708	0.685	0.667
+30	0.736	0.711	0.689	0.670
+40	0.740	0.715	0.693	0.674
+50	0.743	0.719	0.697	0.677
+60	0.746	0.722	0.700	0.680
+70	0.750	0.726	0.704	0.684

* Dickinson, H. C., E. F. Mueller, and E. B. George, "Specific Heat of Some Calcium Chloride Solutions," *Bur. Standards Bull.* 6, pp. 379–408, 1910 (*Sci. Paper* S-135).

The results of a series of observations on a sample of calcium chloride give the following formulas for the specific heat:

Density, 1.260, $s = 0.666 + 0.00064t$ (from −35 to +15° C.)
Density, 1.200, $s = 0.708 + 0.00064t$ (from −20 to +15° C.)
Density, 1.140, $s = 0.772 + 0.00064t$ (from −10 to +15° C.)
Density, 1.070, $s = 0.869 + 0.00057t$ (from 0 to +15° C.)

where s is the specific heat and t is the temperature. All densities are referred to a temperature of 20° C. in terms of water at 4° C.

In using the specific-heat values for the desired temperature, it is noted that for a given density the equation is linear; because of this, the mean specific heat must be used for the range of temperature.

The following freezing points for chemically pure calcium chloride were obtained by test:

Density	Freezing Temperatures, Degrees Centigrade
1.12	− 9
1.14	−13
1.16	−16
1.18	−20
1.20	−24
1.22	−29
1.24	−34
1.26	−40

TABLE XIII.—DATA OF TYPICAL DOUBLE-ACTING MEDIUM-SPEED AMMONIA COMPRESSORS

Size of compressor	Speed, revolutions per minute	Displacement at listed speed
5 × 8	125	38,580
6½ × 10	120	78,000
7¾ × 11	110	112,500
9 × 12	100	149,140
10 × 18	85	236,000
12 × 20	85	376,500
12 × 24	82	436,000
13½ × 24	80	536,080
15 × 30	70	726,000
16½ × 30	70	884,000
18 × 30	70	1,040,000
19 × 36	70	1,396,000

TABLE XIV.—SURFACE COEFFICIENTS FOR VARIOUS MATERIALS

Kind of material	Coefficient for still air, B.t.u. per square foot per hour per degree Fahrenheit
Brick wall	1.4
Concrete, 1-2-4 mixture	1.50
Wood (fir, one side finished)	1.40
Cork board	1.25
Magnesia board	1.45
Tile, plaster on both sides	1.10

For moving air having a velocity of about 15 miles per hour, take the value of the surface coefficient to be equal to three times the above coefficient for still air.

TABLE XV.—HEAT-TRANSMISSION COEFFICIENTS OF CORK BOARD, GRANULATED CORK, AND MILL SHAVINGS

B.t.u. per 24 hour per square foot per degree Fahrenheit	Thickness of cork board, inches	Thickness of granulated cork, inches	Thickness of mill shavings, inches
10	1	2	2¾
5	2	4	5½
3½	3	6	8
3	4	8	10¾
2½	5	10	13½
2	6	12	16
1¼	8	16	21½

TABLE XVI.—HEAT TRANSMISSION THROUGH PIPES UNDER VARIOUS CONDITIONS

½ B.t.u. per square foot per hour per degree Fahrenheit	Direct expansion to still air
B.t.u. per square foot per hour per degree Fahrenheit	Direct expansion to forced air
B.t.u. per square foot per hour per degree Fahrenheit	Flooded system to still air
½ B.t.u. per square foot per hour per degree Fahrenheit	Flooded system to forced air
4½ B.t.u. per square foot per hour per degree Fahrenheit	Brine piping to still air
½ B.t.u. per square foot per hour per degree Fahrenheit	Brine piping to forced air
12 B.t.u. per square foot per hour per degree Fahrenheit.	Direct expansion to liquid (submerged)
20 B.t.u. per square foot per hour per degree Fahrenheit.	Flooded system to liquid (submerged)

TABLE XVII.—CONDUCTIVITY AND DENSITY OF VARIOUS INSULATING MATERIALS

U. S. Bureau of Standards

Material	Thermal conductivity B.t.u. per square foot, per inch, per hour, per degree Fahrenheit	Density pounds per cubic foot	Description of material
Air	0.175	0.08	Ideal air space.
Air cell, ½ inch	0.458	8.80	Asbestos paper and air spaces.
Air cell, 1 inch	0.500	8.80	Asbestos paper and air spaces.
Asbestos mill board	0.830	61.0	Pressed asbestos.
Asbestos wood	3.700	123.0	Asbestos and cement.
Balsa wood	0.350	7.5	Light and soft across grain.
Calorax	0.221	4.0	Fluffy, finely divided mineral matter.
Cork	0.337	5.3	Granulated ⅛–³⁄₁₆ inch.
Cork	0.330	10.0	Regranulated ¹⁄₁₆–⅛ inch.
Corkboard	0.279	6.9	No artificial binder-low density.
Corkboard	0.308	11.3	No artificial binder-high density.
Cotton wool	0.292	......	Loosely packed.
Fibrofelt	0.329	11.3	Felted vegetable fibers.
Fire felt wool	0.625	43.0	Asbestos sheet coated with cement.
Fire felt sheet	0.583	26.0	Soft, flexible asbestos sheet.
Flaxlinum	0.329	11.3	Felted vegetable fibers.
Hair felt	0.246	17.0	
Hard maple wood	1.125	44.0	Across grain.
Infusorial earth	0.583	43.0	Natural blocks.
Insulite	0.296	11.9	Pressed wool pulp-rigid.
Kapok	0.238	0.88	Vegetable fiber-loosely packed.
Keystone hair	0.271	19.0	Hair felt combined with building paper.
Linofelt	0.300	11.3	Vegetable fiber combined with paper.
Lithboard	0.379	12.5	Mineral wool and vegetable fibers.
Mineral wool	0.275	12.5	Medium packed.
Mineral wool	0.288	18.0	Felted in blocks.
Oak wood	1.000	38.0	Across grain.
Planer shavings	0.417	8.8	Various.
Pulp board	0.458	......	Stiff pasteboard.
Pure wool	0.263	5.0	
Rock cork	0.346	21.0	Mineral wool and binder—rigid.
Slag wool	0.750	15.0	
Tar roofing	0.707	55	
Virginia pine wood	0.958	34	Across grain.
White pine wood	0.791	32	Across grain.
Wool felt	0.363	21	Flexible paper stock.

TABLE XVIII.—COEFFICIENT OF CONDUCTIVITY OF BUILDING MATERIALS

	Coefficient
Brickwork	5.0
Concrete	5.3 (average)
Wood (fir, 7/8 inch thick)	1.0
Asbestos (sheets or boards)	0.3 to 0.5
Glass (0.085 inch thick)	24.3
Double window (1/2 inch air space)	1.10
2-inch hollow tile (plastered)	1.0
4-inch hollow tile (plastered)	0.6
Mortar	8.0

The figures are the B.t.u. per square foot, per inch thick (unless the thickness is mentioned) per degree difference of temperature per hour.

TABLE XIX.—THERMAL CONDUCTIVITY OF EARTHY MATERIALS
Tests made by the Food Investigation Board, 1921*

Material	Gram-calories per square centimeter per second, for 1 centimeter thickness and for 1° C. difference in temperature	B.t.us. per square foot per hour for 1 inch thickness and for 1° F. difference in temperature	Temperature range	Mean temperature of the insulating material
Diatomaceous earth	0.000193	0.560	9 to 49° F.	30° F.
Diatomaceous brick	0.000223	0.647	4 to 80° F.	42° F.
Concrete block (used for construction work)	0.0028	8.2	60 to 115° F.	88° F.

* *Special Rept.* 5, H.M. Stationery Office, London.

TABLE XX.—STANDARD DIMENSIONS EXTRA HEAVY PIPE

Diameter			Thickness, inches	Circumference		Length of pipe per square foot		Transverse areas			Nominal weight, pounds per foot	Number of threads, per inch
Nominal, inches	External, inches	Internal, inches		External, inches	Internal, inches	External surface, square inches	Internal surface, square inches	External, square inches	Internal, square inches	Metal, square inches		
1/8	0.405	0.205	0.100	1.272	0.644	9.44	18.63	0.129	0.033	0.096	0.29	27
1/4	0.540	0.294	0.123	1.696	0.924	7.07	12.99	0.229	0.068	0.161	0.54	18
3/8	0.675	0.421	0.127	2.121	1.323	5.66	9.07	0.358	0.139	0.219	0.74	18
1/2	0.840	0.542	0.149	2.639	1.703	4.55	7.05	0.554	0.231	0.323	1.09	14
3/4	1.050	0.736	0.157	3.299	2.312	3.64	5.11	0.866	0.425	0.441	1.39	14
1	1.315	0.951	0.182	4.131	2.988	2.90	4.02	1.358	0.710	0.648	2.17	11 1/2
1 1/4	1.660	1.272	0.194	5.215	3.996	2.30	3.00	2.164	1.271	0.893	3.00	11 1/2
1 1/2	1.900	1.494	0.203	5.969	4.694	2.01	2.56	2.835	1.753	1.082	3.63	11 1/2
2	2.375	1.933	0.221	7.461	6.073	1.61	1.97	4.430	2.935	1.495	5.02	11 1/2
2 1/2	2.875	2.315	0.280	9.032	7.273	1.33	1.65	6.492	4.209	2.283	7.67	8
3	3.500	2.892	0.304	10.996	9.086	1.09	1.33	9.621	6.569	3.052	10.25	8
3 1/2	4.000	3.358	0.321	12.566	10.549	0.955	1.14	12.566	8.856	3.710	12.47	8
4	4.500	3.818	0.341	14.137	11.995	0.849	1.00	15.904	11.449	4.455	14.97	8
4 1/2	5.000	4.280	0.360	15.708	13.446	0.764	0.893	19.635	14.387	5.248	18.22	8
5	5.563	4.813	0.375	17.477	15.120	0.687	0.793	24.306	18.193	6.113	20.54	8
6	6.625	5.751	0.437	20.813	18.067	0.577	0.664	34.472	25.976	8.496	28.58	8
7	7.625	6.625	0.500	23.955	20.813	0.501	0.598	45.664	34.472	11.192	37.67	8
8	8.625	7.625	0.500	27.096	23.955	0.443	0.502	58.426	45.664	12.762	43.00	8
9	9.625	8.625	0.500	30.238	27.096	0.397	0.443	72.760	58.426	14.334	48.25	8
10	10.750	9.750	0.500	33.772	30.631	0.355	0.399	90.763	74.662	16.101	54.25	8
12	12.750	11.750	0.500	40.055	36.914	0.299	0.325	127.68	108.43	19.25	65.00	8

Table XXI.—Comparative Capacities of Pipes of Standard Sizes, Showing the Number of Times the Area of One Pipe Is Contained in That of a Larger Size

	⅛	¼	⅜	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	7	8	9	10
⅛	1																		
¼	1 9	1																	
⅜	3 3	1 8	1																
½	5 3	2 9	1 5	1															
¾	9 3	5 2	2 7	1 7	1														
1	14 7	7 7	4 1	2 6	1 6	1													
1¼	26 2	14 3	7 8	4 9	2 8	1 6	1												
1½	35.6	19 5	10 6	6 6	3 8	2 4	1 3	1											
2	58 6	32 2	17 5	11 0	6 9	8 9	2 2	1 6	1										
2½	83.6	44 6	25 0	15 7	9 0	5 5	3 2	2 3	1 4	1									
3	129 1	71 0	38 5	24 2	13 8	8 5	5 0	3 6	2 2	1 5	1								
3½	172 8	95 0	51 6	32.4	18 5	11 4	6 6	4 8	2 9	2 0	1 3	1							
4	222.5	122 3	66 4	41 7	23 8	14 7	8 5	6 2	3 8	2 6	1 7	1 3	1						
5	349 4	192 0	104 3	65 6	37 4	23 1	13 3	9 8	5 9	4 1	2 7	2 0	1 5	1					
6	505 0	277 5	150 7	94 7	54 1	33 4	10 3	14 1	8 6	6 0	3 9	2 9	2 2	1 4	1				
7	677 2	372 1	202.2	127 0	71 3	44 9	25 9	19 0	11 5	8 1	5 2	3 9	3 0	1 9	1 3	1			
8	878 3	480 7	261 2	164 1	93 8	58 0	34 5	24 5	14 9	10 4	6 7	5 0	3 9	2 5	1 7	1 3	1		
9	1112 4	611 2	332 1	208 7	179 3	73 4	42 7	31 2	19 0	13 3	8 7	6 4	5 0	3 1	2 2	1 6	1 2	1	
10	1378.3	757 3	411 5	258 6	147 8	91 3	52 7	38 6	23 5	16 5	10 6	7 0	6 2	3 9	2 7	2 0	1 5	1 2	1

Table XXII.—Wrought Iron Welded Steam, Gas, and Water Pipe

Table of Standard Dimensions

Diameter			Thickness Inches	Circumference		Transverse Areas			Length of Pipe per Sq. Foot of		Length of Pipe Cont'g One Cu. Foot	Weight per Ft. of Length Pounds	No. of Threads per Inch of Screw	Contents in Gals. per Foot of Length	Weight of Water per Foot of Length Pounds
Nom. Inter'l In.	Actual External Inches	Actual Internal Inches		External Inches	Internal Inches	External Sq. Inch	Internal Sq. Inch	Metal Sq. Inch	External Surface Feet	Internal Surface Feet					
⅛	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513.	.241	27	.0006	.005
¼	.54	.364	.088	1.696	1 144	.229	.1041	.1249	7.075	10.49	1383.3	.42	18	0026	021
⅜	.675	.494	.091	2,121	1.552	.358	.1917	.1663	5.657	7.73	751 2	.559	18	.0057	047
½	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14	0102	.085
¾	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3 637	4 635	270.	1 115	14	.0230	.190
1	1.315	1.048	.134	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.668	11½	.0408	349
1¼	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2 768	96 25	2 244	11½	.0638	.527
1½	1.9	1.611	.145	5 969	5.061	2.835	2.038	797	2 01	2.371	70.66	2 678	11½	0918	.760
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42 91	3 609	11½	1632	1.356
2½	2.875	2.468	.204	9.032	7.753	6.492	4.784	1 708	1.328	1.547	30 1	5.739	8	2550	2.116
3	3.5	3.067	.217	10 996	9.636	9.621	7 388	2 243	1.091	1 245	19.5	7 536	8	.3673	3 049
3½	4.	3.548	.226	12 566	11 146	12 566	9 887	2.679	.955	1 077	14.57	9 001	8	4998	4 155
4	4.5	4.026	.237	14 137	12 648	15 904	12.73	3 174	.849	949	11.31	10 665	8	6528	5 405
4½	5.	4.508	.246	15.708	14.162	19 635	15 961	3.674	.764	848	9 02	12.34	8	8263	6 851
5	5 563	5.045	.259	17.477	15.849	24 306	19 99	4.316	.687	757	7.2	14.502	8	1 020	8 500
6	6.625	6.065	.28	20.813	19.054	34 472	28.888	5.584	.577	63	4.98	18.762	8	1.469	12.312
7	7.625	7 023	.301	23 955	22.063	45.664	38.738	6.926	.501	.544	3 72	23.271	8	1 999	16 662
8	8.625	7.982	.322	27 096	25.076	58 426	50.04	8.386	.443	.478	2 88	28.177	8	2 611	21.750
9	9.625	8.937	.344	30 238	28.076	72.76	62 73	10 03	.397	427	2.29	33 701	8	3 300	27 500
10	10.75	10.019	.366	33.772	31 477	90 763	78.839	11.924	.355	.382	1 82	40.065	8	4 081	34 000
11	12.	11.25	.375	37.699	35 343	113.098	99 402	13.696	318	.339	1 456	45 95	8	5 163	43 000
12	12.75	12.	.375	40 055	37 7	127 677	113 098	14.579	299	319	1 27	48.985	8	5.875	48 930

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